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DAVIDSON LABORATORY

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March 1985

DEVELOPMENT OF WATERJET PROPULSION UNIT

by

John K. Roper

Prepared for

Code 112
David W. Taylor
Naval Ship Research and Development Center

Under
Office of Naval Research
Contract N00014-80-D-0890
Delivery Order 4, Item 1

(DL Project 4982/134)

STEVENS INSTITUTE
OF TECHNOLOGY

CASTLE POINT STATION
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

An axial-flow pump was designed to be used in an LVTP-7A1 amphibious vehicle; it shows significant advantages over the existing waterjet unit in efficiency, thrust output and system weight. Also, two pumps were designed to provide cavitation-free performance at propulsive coefficients in the region of 40 to 50 percent for a proposed high-speed amphibious vehicle designed to run at a water speed of 20 mph. State-of-the-art composite material technology was used wherever possible to reduce weight.

STEVENS INSTITUTE OF TECHNOLOGY
DAVIDSON LABORATORY
Castle Point Station, Hoboken, New Jersey 07030

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DEVELOPMENT OF WATERJET PROPULSION UNIT

By

John K. Roper

for

David W. Taylor Naval Ship Research and Development Center
Code 1120

under

Office of Naval Research
Contract N00014-80-D-0890
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DL Project 4982/134

APPROVED: ____________________
Daniel Savitsky
Director

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SUMMARY

Designs of three waterjet propulsion systems have been developed for application to amphibious vehicles. The first system was an axial-flow pump designed to be used in an existing amphibious vehicle, an LVTP-7A1 and it shows significant advantages over the existing waterjet unit in efficiency, thrust output and system weight.

The other two systems were to be used to propel a proposed high-speed amphibious vehicle. These pumps were designed to provide cavitation-free performance at propulsive coefficients in the region of 40 to 45 percent at a vehicle water speed of 20 mph. State-of-the-art composite material technology was used wherever possible to reduce weight.

INTRODUCTION

The U. S. Marine Corps plans to improve the mobility of amphibious vehicles. One aspect which requires attention is the need to increase the efficiency of existing waterjet propulsion units, along with improving the durability and reducing the cost of these units.

Existing waterjet installations in typical amphibious vehicles have low efficiencites due to numerous design constraints associated with their present stern locations. One of the contributors to their relatively poor performance is the location of the water intakes to an area which is seriously obstructed by the tracks. Although this is but one influence on total performance, it is desirable to define the sources of blockage, interference, ventilation, etc., in the intake area of presently installed waterjets and to then use these results to recommend suitable design changes which will ameliorate the undesirable effects.

It is also appropriate to consider a redesign of the present waterjet units using modern high strength plastic materials being developed by AMRAC at the Watertown Arsenal. These plastics should have a high resistance to erosion by sand or debris which can pass through the impeller and hence result in a more durable and potentially less costly propulsion unit.

This design effort proceeded through the following phases:

1. Feasibility study of composite plastic waterjet propulsion unit.
2. Design study of a waterjet unit with improved propulsion efficiency for:
   a. An existing slow-speed-in-water amphibious vehicle
   b. A proposed high-speed-in-water vehicle

FEASIBILITY STUDY OF COMPOSITE PLASTIC WATERJET UNIT

In order to evaluate the practicality of constructing a composite plastic waterjet pump and to obtain expert advice on likely materials and fabrication methods, discussions were held with Mr. A. Alisio of the Army Materials Research Laboratory of Watertown, MA and Mr. A. Macander of the Naval Research & Development Center at Annapolis, MD.

There is no doubt that the concept is well within the state-of-the-art. Indeed, similar components such as composite plastic pump casings, impellers and large valves are in production and are used extensively in many industries. The question is whether the tooling costs, which are likely to be high, can be justified by the small number of units to be produced. This question can only be answered by obtaining cost estimates from manufacturers for specific components.

On the basis of advice received so far, the propeller duct, Figure 1, would be layed up of "pre-preg" fabric over a male mold or perhaps filament-wound. The propeller, support strut, rudder bearings and other small parts would be injection or transfer molded.

While composites of carbon fiber with polyurethane resin have been recommended because of their stiffness and abrasion resistance, it appears that components similar to those shown in Figure 1, have been fabricated successfully using glass fibers with a variety of other resins such as acetal, polycarbonate, epoxy, etc., and that these composites should be investigated further.

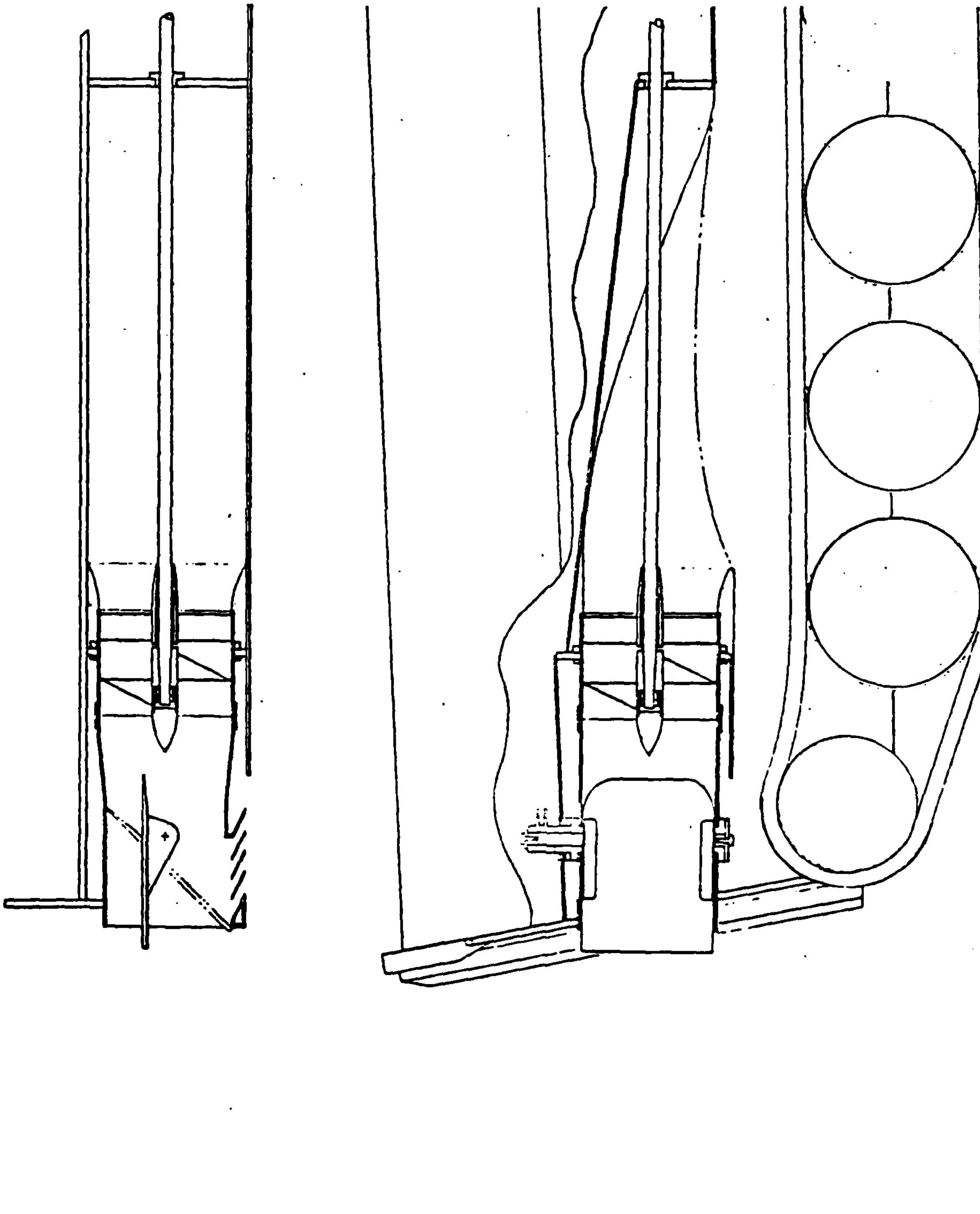


FIGURE 1 PROPOSED WATERJET PROPULSION SYSTEM FOR LVTP-7A1

DESIGN OF WATERJET PROPULSION SYSTEM FOR LVTP-7A1 AMPHIBIOUS VEHICLE

The general objective was to design an axial flow pump, two of which would generate sufficient thrust to propel an existing amphibious vehicle - an LVTP-7A1 - at a cruise speed of 8 mph with a propulsive efficiency higher than that of the existing propulsor.

Figure 1 shows the configuration of the proposed propulsion system in the aft end of an LVTP-7A1. A 20-inch diameter impeller is to be housed in a horizontal cylindrical duct, and water would be drawn from the track well through a horizontal rectangular inlet. There is a transition from the 8 sq ft horizintal inlet area to a 22-inch by 24-inch vertical opening, and thence to the 20-inch nominal diameter of the cylindrical duct. A rudder and a reversing elbow are located at the duct outlet.

For purposes of calculating system performance, the impeller is assumed to be a marine screw propeller with wide tips, having a blade area ratio that is commercially available. Reference 1 presents open water characteristics of such a screw propeller in an axial cylinder. Figure 2 shows charts adapted from Reference 1 (Figures 28 and 29, respectively).

A matrix of design calculations involving the primary variables of pump input power, pump flow rate, and vehicle speed was completed to determine:

(a) The pump head rise which can be produced by a given pump input horsepower for a range of flow rates.

(b) The pump head rise at which cavitation begins to affect pump performance at given vehicle speeds for a range of flow rates.

(c) The pump head rise required to produce a given flow rate through the duct system over a range of vehicle speeds.

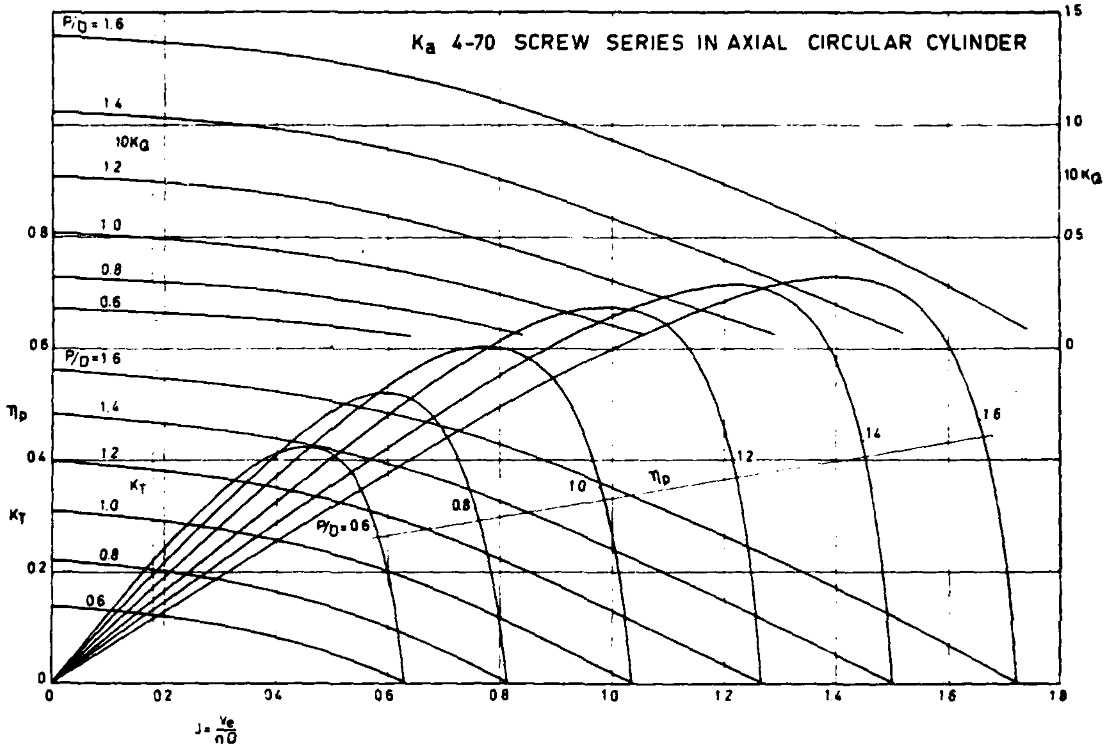


Fig. 28 Results of open-water tests with Ka 4-70 screw series in an axial cylinder

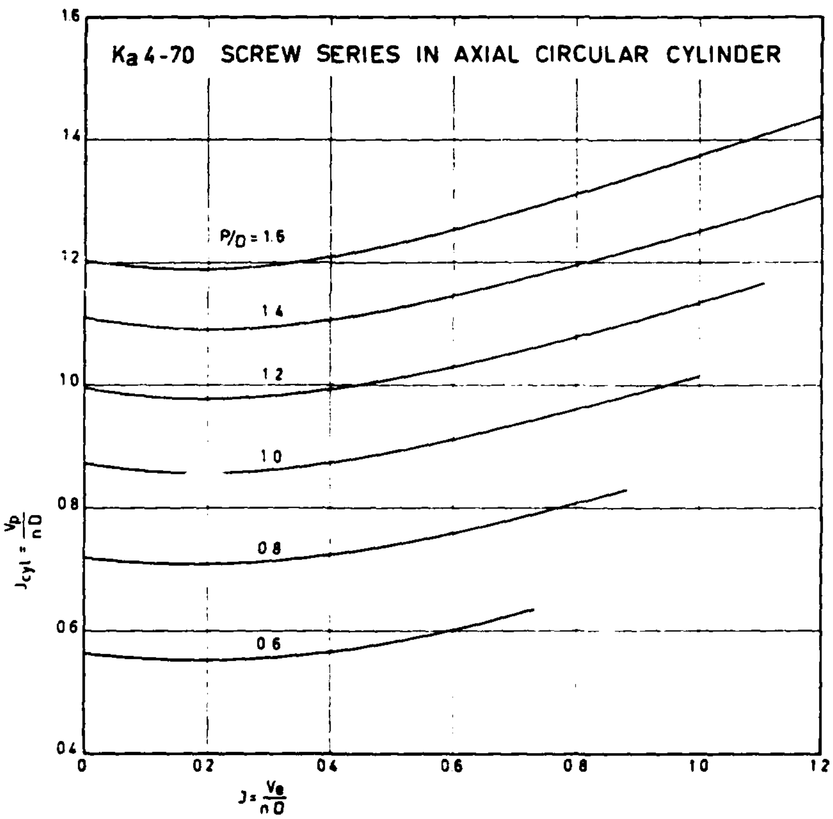


Fig. 29 Relation between velocity of "screw + cylinder" combination and velocity in cylinder

FIGURE 2 "SCREW-IN-CYLINDER" DIAGRAMS (REFERENCE 1)

Appendix A presents the details of these calculations.

Figure 3 is a chart of pump head H_p versus flow rate Q in the form in two families of curves showing the results of calculations (b) and (c) above, with vehicle speed as a parameter. Equilibrium flow rate and pump head rise were determined for a given vehicle speed at the intersection of the Required H_p and Available H_p curves for that speed.

Figure 4 is a chart of H_p versus Q showing results of calculations (a) and (c) above, with input power SHP and vehicle speed V_o, respectively, as parameters of the two families of curves. Entering Figure 4 with the equilibrium flow rate for a given speed from Figure 3, permits the determination of input power SHP required at equilibrium.

From the equilibrium flow rate Q cu ft/sec and the exit duct area, the exit jet velocity V_J was calculated. Jet thrust T, the time rate of change of fluid momentum, was then determined:

$$T = \rho Q(V_J - V_o)$$

The ratio of output power, $TV_o/550$, to input SHP was then the propulsive coefficient, P.C.

Having determined hydrodynamic loads, required input power, and propeller operating conditions, a structural analysis of propeller, shaft, rudder and duct was performed to determine required sizes. Finally, weight estimates were made assuming (a) aluminum construction and (b) composite materials construction of the waterjet system.

From known characteristics of the existing waterjet system in the LVTP-7A1 amphibious vehicle, the following comparison was developed:

		Existing System	Proposed System
At 8 mph:	Thrust, lb	2369	2846
	Flow, gpm	14020	33346
	P. C.	.25	.30
At 0 mph:	Thrust, lb	3025	4278
Dry Weight, lb		435	284 (aluminum) 197 (composites)

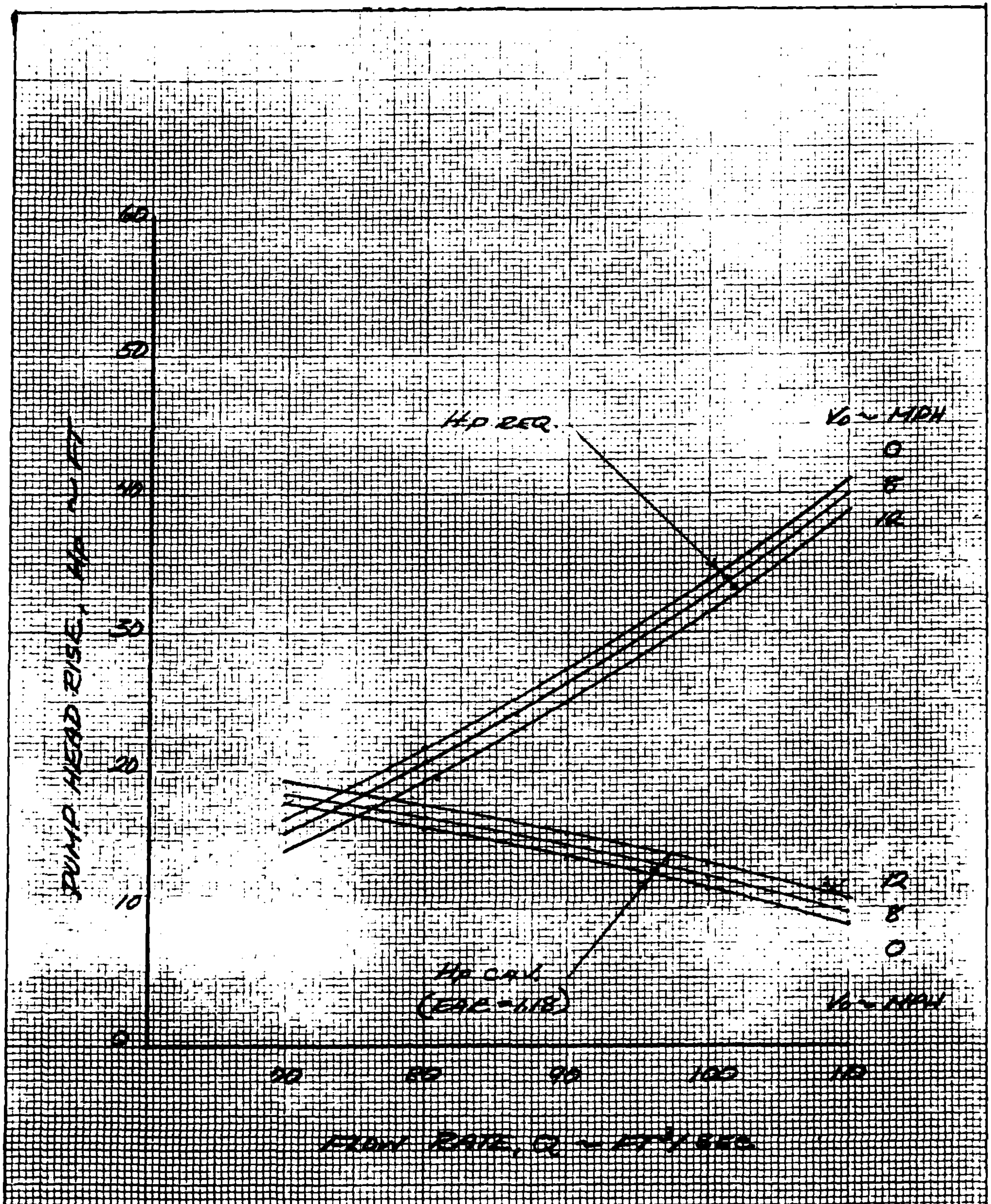


FIGURE 3 CAVITATION LIMITS

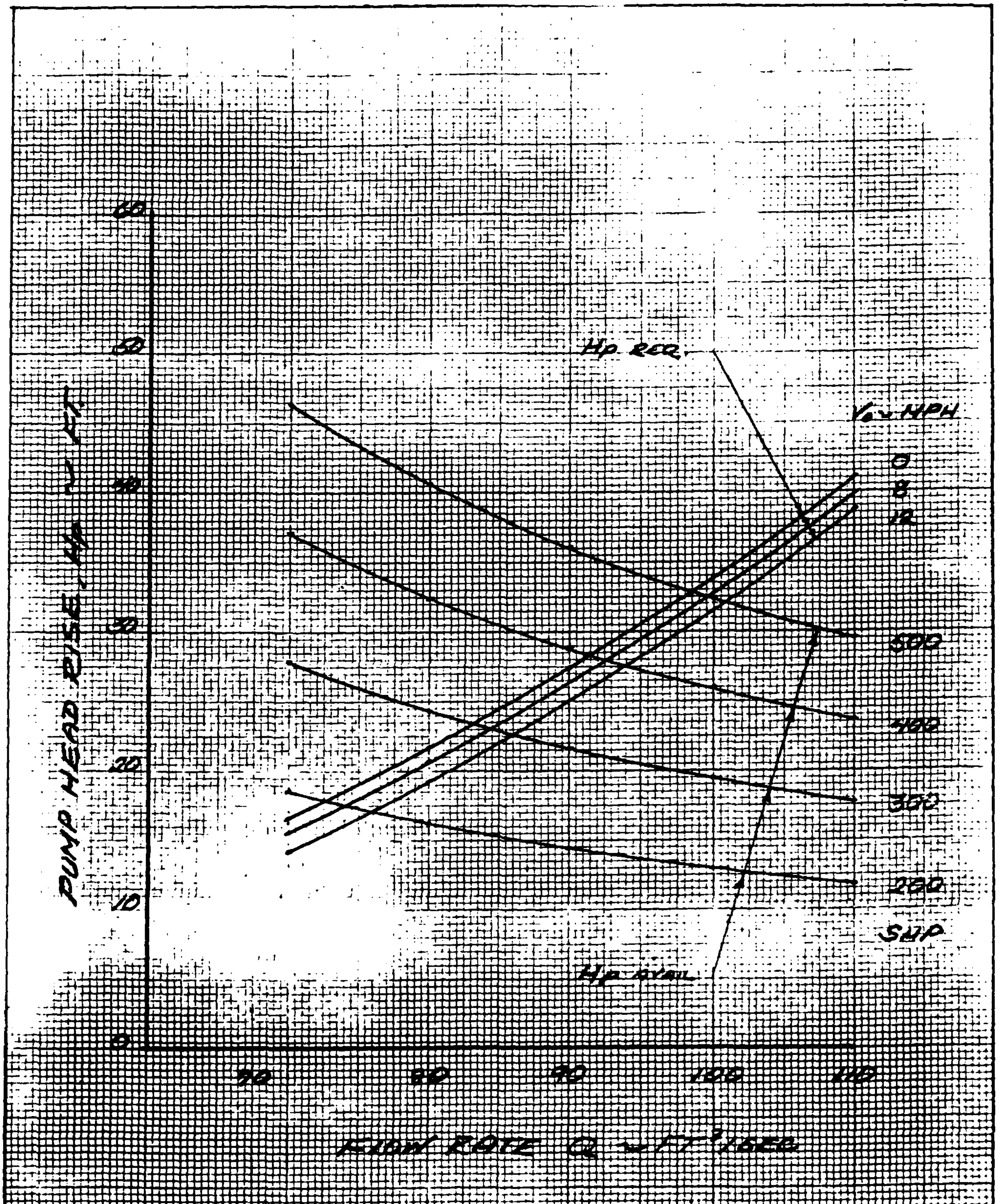


FIGURE 4 POWER LIMITS

DESIGN OF PROPULSION PUMP SYSTEM FOR HIGH-SPEED AMPHIBIAN

The general objective was to design an axial flow pump suitable for use in a multiple unit waterjet propulsion system in a high-speed amphibious vehicle to achieve a 20 mph speed.

Figure 5 shows an elevation sketch of such a unit which would draw water through a 42 inch x 20 inch rectangular port in the flat bottom of the amphibian. The flow then passes through a 24 inch x 20 inch inlet to a short transition and finally through a cylindrical duct with a nominal diameter of 20 inches in which the 20 inch diameter pump impeller is located.

For purposes of calculating system performance, the pump impeller was assumed to be a marine screw propeller with wide tips and a largest commercially-available blade area ratio. Appendix B presents details of calculations of:

(a) Pump head rise versus flow rate for selected input powers, i.e., power-limited head rise.

(b) Pump head rise versus flow rate for selected vehicle speeds, such that pump performance is not affected by cavitation, i.e., cavitation-limited head rise.

(c) Pump head rise versus flow rate for selected vehicle speeds required to overcome system head losses.

These calculations made use of propeller performance data in cylindrical ducts, Figure 2 (from Reference 1), and a curve of inlet ram pressure recovery ratio in Appendix B, page B-22.

Equilibrium flow rate and pump head rise were determined, for a given vehicle speed, at the intersection of the curve of cavitation-limited head rise with the corresponding curve of head rise required to overcome system head losses.

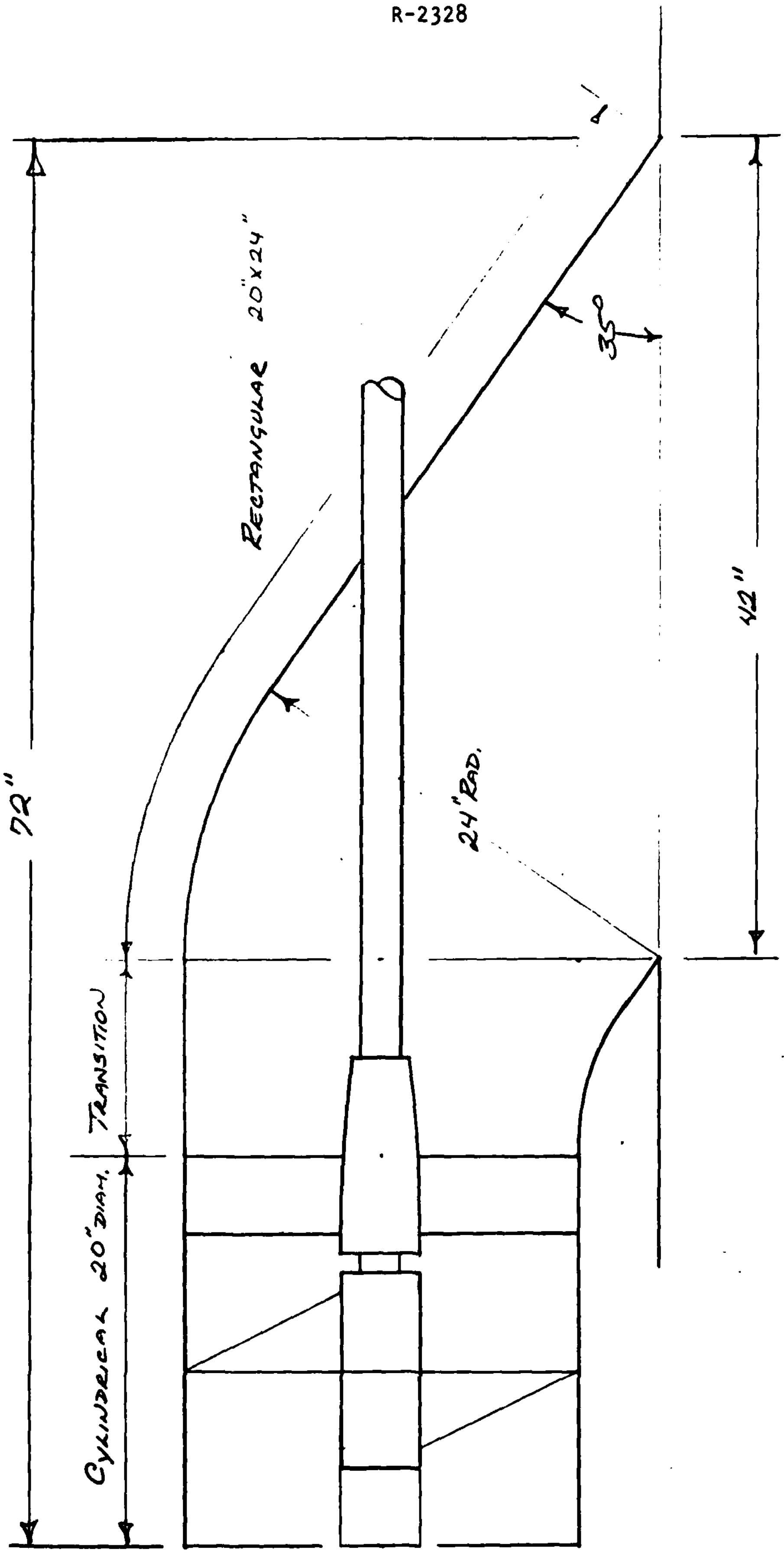


FIGURE 5 SKETCH OF 20 INCH DIAMETER PUMP

Knowing the equilibrium head rise and flow rate Q at a given vehicle speed, calculation procedure (a) was used to determine input power SHP. Jet thrust T was calculated after finding jet velocity from Q and the exit duct area. Finally, at a given vehicle speed, the ratio of thrust horsepower output to input SHP gave propulsive coefficient P.C.

Having determined hydrodynamic loads, required input power and propeller operating conditions for cavitation-free performance with an area ratio of 1.0, structural analyses of propeller, shafting and casing were performed to determine required sizes. Then weight estimates were made assuming (a) an aluminum casing, and (b) a composite-materials casing; aluminum alloy shafting and Ni-Al bronze propeller were used in each case. Selected performance characteristics were:

At zero mph:	Thrust	4507 lb
At 20 mph	Thrust	2365 lb
	Flow	40,080 gpm
	Input SHP	284 hp
	P.C.	.445

Composite construction of the casing reduced the dry weight of the waterjet system to 169 lb from a 239 lb weight for aluminum construction.

DESIGN OF 15-INCH DIAMETER PROPULSION PUMP SYSTEM FOR HIGH-SPEED AMPHIBIAN

The general objective was to design an axial flow pump suitable for installation in a multiple unit waterjet propulsion system in a high-speed amphibious vehicle to achieve a 20 mph speed.

Figure 6 is an elevation sketch of the proposed unit which would draw water through a 31½ inch x 15 inch rectangular port in the flat bottom of the amphibian. The flow then passes through an 18 inch x 15 inch inlet to a short transition and finally through a cylindrical duct with a nominal diameter of 15 inches in which a 15 inch diameter pump impeller is located.

This 15 inch diameter impeller is to provide at least the same propulsive thrust as the 20 inch diameter propeller described in the previous section because the same vehicle is involved. To meet this loading requirement requires a significant increase in impeller blade area ratio if cavitation is to be avoided. Thus, the calculations in Appendix C include consideration of projected area ratios, PAR = 1.0, 1.5, 2.0, 2.5 and 3.0. By contrast the largest commercially available PAR is about 1.0.

Assuming a projected area ratio of 3.0 as an upper limit for extended cavitation-free operation, structural analyses of propeller, shaft and ducting were performed to determine required sizes. Weight estimates were made assuming (a) aluminum casing, and (b) a composite materials casing; Acquamet 22 shafting and Ni-Al bronze impeller were used in each case. Selected performance characteristics were:

At zero mph:	Thrust, lb	5,403
At 20 mph:	Thrust, lb	3,703
	Flow, gpm	30,120
	Input SHP	462
	P.C.	.428

Composite construction of the casing reduced the dry weight of the waterjet system to 134 lb from a 167 lb weight for aluminum construction.

The next step in design would require consideration of available engine powers, and the thrust needed to overcome vehicle drag in order to settle on a practical area ratio for the impeller.

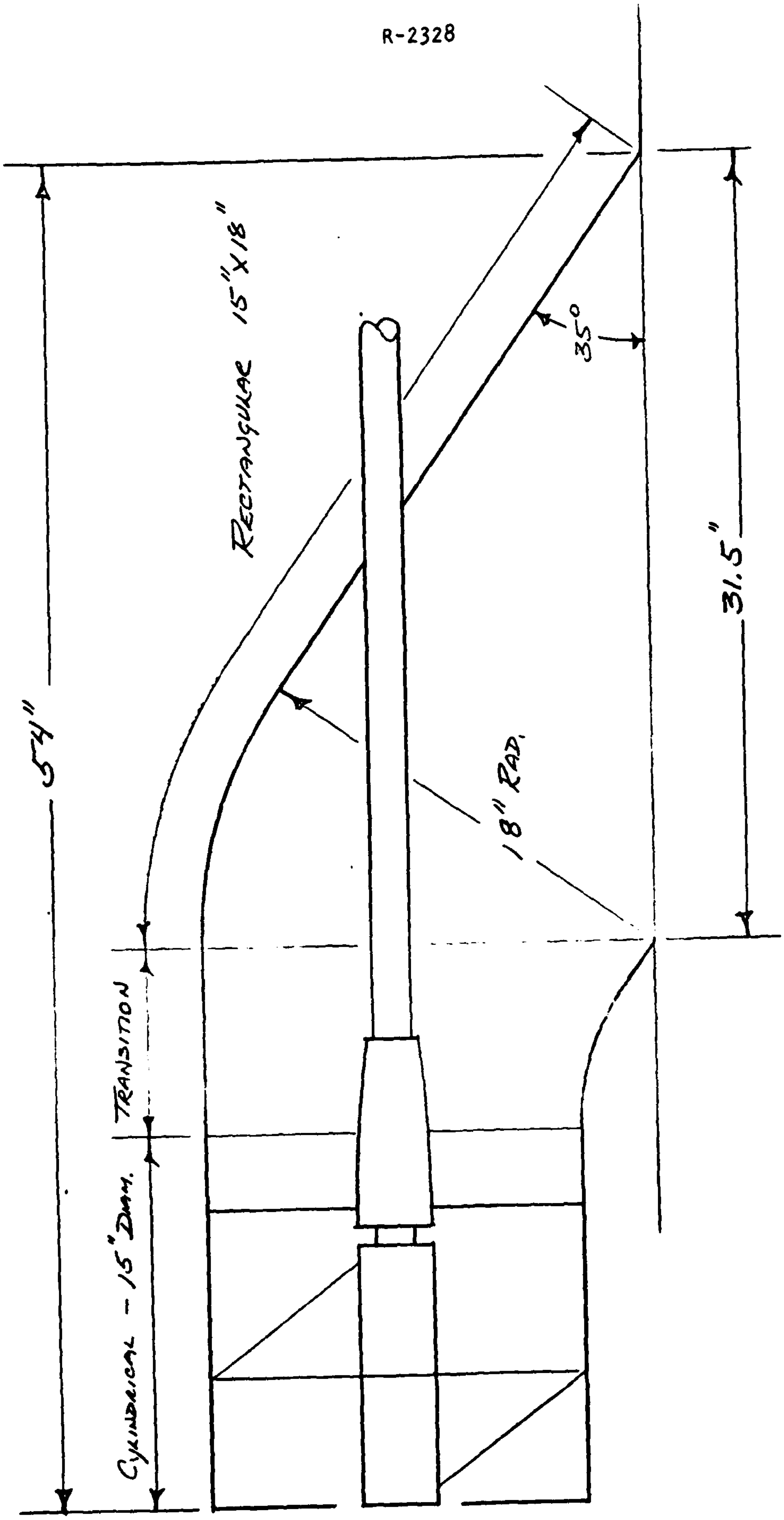


FIGURE 6 SKETCH OF 15 INCH DIAMETER PUMP

REFERENCES

1. van Manen, J.D. and Oosterveld, M.W.C., "Analysis of Ducted-Propeller Design," Trans. SNAME, Vol. 74, 1966.

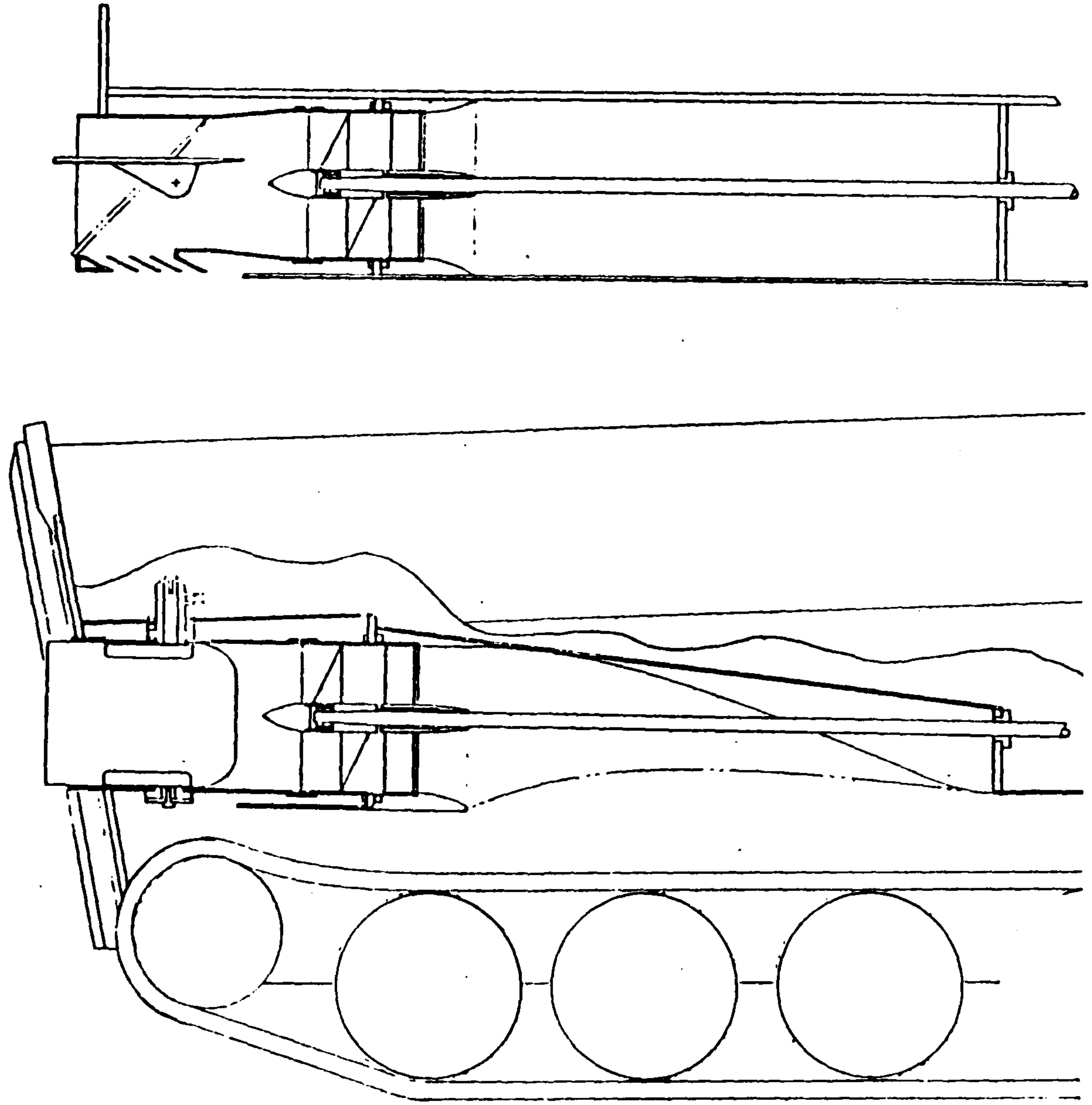
2. "Flow of Fluids Through Valves, Fittings and Pipe," Crane Company Technical Paper 410.

3. Conolly, J.E., "Strength of Propellers," Trans. Royal Institution of Naval Architects, 1960.

APPENDIX A

OBJECTIVES:

- Design a replacement propulsion system for an LVTP-7A1 amphibious vehicle.
- Determine performance, weight and dimensional characteristics of propulsion system.
- Use simple "propeller-in-tube" approach
- Limit blade area ratio to that available is existing propeller series.
- Investigate use of composite materials.



PROPOSED WATERJET PROPULSION SYSTEM FOR LVTP-7A1

PERFORMANCE CHARACTERISTICS

- POWER LIMITS
- CAVITATION LIMITS
- SYSTEM PERFORMANCE
- REVERSE OPERATION
- D[illegible]

[illegible]

CALCULATION NOTES

D = [illegible] = 20" [illegible]

A_p = [illegible] AREA [illegible] $.785\left[(1.667)^2-(.333)^2\right] = 2.0942\ FT^2$

A_I = INLET [illegible] $= 3.667\ FT^2$

J_{cyl} = ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE

J_e = ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE

$$\frac{J_{cyl}}{J_e} = \frac{A_I}{A_p} = \frac{3.667}{2.0942} = 1.7510$$

P/D = PROP. PITCH DIAM. RATIO

$J_{cyl} = f\left(P/D, \frac{J_{cyl}}{J_e}\right) = .895$ (VON LAMMEREN – FIG. 1)

$J_e = J_{cyl}/\left(\frac{J_{cyl}}{J_e}\right) = .895/1.7510 = .511$

η_o = PROPULSION EFFICIENCY OF PROP-TUBE COMB. = $f(J_e, P/D)$ (FIG. 2)

η_{RR} = PROP. RELATIVE ROTATIVE [illegible] (G[illegible])

η_p = PUMP EFFICIENCY = [illegible]

SHP = PUMP INPUT [illegible]

Q = FLOW [illegible]

H_{FAN} = PUMP [illegible] $\dfrac{550\ SHP\ \eta_p}{\rho g Q}$

POWER LIMIT

CALCULATION

D	A_P	A_I	J_{cyl}/J_e	P/D	J_{cyl}	J_e	η_o	η_{RR}	η_P	SHP	Q	H_{pavail}
1.667	2.0942	3.667	1.751	1.00	.895	.5111	.46	.95	.76	500	70	46.36
											80	40.57
											90	36.06
											100	32.45
											110	29.50
										400	70	37.09
											80	32.45
											90	28.85
											100	25.96
											110	23.60
										300	70	27.82
											80	24.34
											90	21.64
											100	19.47
											110	17.70
										200	70	18.54
											80	16.23
											90	14.42
											100	12.98
											110	11.80

CAVITATION LIMIT

CALCULATION No. 2

D = PROP. DIAM. = 20" = 1.667'

A_p = PROP. DISC AREA = .785[(1.667)² − (.333)²] = 2.0942 FT²

A_I = INLET AREA (.785)(...) = 3.667 FT²

V_{cyl} = ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE

V_e = ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE

$$\frac{V_{cyl}}{V_e} = \frac{A_I}{A_p} = \frac{3.667}{2.0942} = 1.7510$$

P/D = PROP. PITCH DIAM. RATIO

$V_{cyl} = f(P/D, \frac{V_{cyl}}{V_e})$ (VON LAMMEREN − FIG. 1)

Q = FLOW RATE ~ FT³/SEC

n = PROP. SPEED = $\dfrac{Q}{A_p V_{cyl} D}$

H_{LI} = INLET HEAD LOSS = .000533 Q^2 (DERIVATION #1)

V_I = INLET VELOCITY = Q/A_I

$V_{.7R(t)}$ = TANGENTIAL VELOCITY AT .7R ...

$V_{.7R}$ = TOTAL VELOCITY AT .7R ... = $\sqrt{V_I^2 + V_{.7R(t)}^2}$

H_{ATM} = HEAD DUE TO ...

H_Z = HEAD DUE TO ELEVATION

H_V = HEAD DUE TO VAPOR PRESS.

CALCULATION NOTE (CONT.)

V_0 = FREE STREAM [illegible]

RPR = RAM PRESS. RECOVERY RATIO = .50 (GUESS)

H_0 = RAM PRESS. RECOVERY = $(RPR)\frac{V_0^2}{2g}$

H_{I_S} = INLET STATIC HEAD (ABOVE VAP. PRESS.) = $H_{ATM} + H_Z - H_V + H_0 - HL_I - \frac{V_I^2}{2g}$

P_{I_S} = INLET STATIC PRESS. (ABOVE VAP. PRESS.) = $\rho g H_{I_S}$

$\sigma_{.7R}$ = LOCAL CAV. NO. AT .7R = $\frac{P_{I_S}}{\frac{\rho}{2} V_{.7R}^2}$

$\tau_{C_{MAX}}$ = PROP. LOAD COEF. AT CAV. LIMIT = $.7\sigma_{.7R}$ (GAWN)

EAR = PROP. EXPANDED AREA RATIO = 1.1[illegible] ([illegible])

PAR = PROP. PROJECTED AREA RATIO = 1.01 ([illegible] # 3)

$H_{P_{CAV}}$ = PUMP HEAD RISE AT CAV. LIMIT = $\frac{(\tau_{C_{MAX}})(PAR)(V_{.7R})^2(\frac{\rho}{2})}{\rho g}$

CAVITATION LIMIT

CALCULATIONS

D	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Q	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Hom	He	Hl	Vo	RPR	Ho	His	Pis	σ	τ	Eff	F	Hreal	
[illegible]	[illegible]	3.2?7	1.751	1.00	.675	70	22.40	2.61	[illegible]	[illegible]	84.31	5.66	33.08	1.00	1.00	0	.50	0	24.81	1598	.2249	.1574	1.18	1.01	19.55	
						80	25.60	3.41	21.62	93.25	96.35	7.37							22.[illegible]	1433	.1546	.1082			15.35	
						90	28.60	4.32	24.7	105.56	[illegible]	7.35							19.41	12?0	.1064	.0745			13.33	
						100	32.01	5.33	27.27	110.35	110.46	11.55							[illegible]	1043	.0719	.0503			11.45	
						110	35.21	6.45	30.??	[illegible]	[illegible]	13.18							[illegible]	[illegible]	.0464	.0325			8.95	
						70										5.88	.50	[illegible]	[illegible]	[illegible]	.2271	.1570			17.73	NOT PLOTTED (CLAPPED)
						80													[illegible]	[illegible]	.1564	.1075			15.94	
						90													[illegible]	[illegible]	.1039	.0755			13.91	
						100													[illegible]	[illegible]	.0731	.0512			11.66	
						110													[illegible]	832	.0474	.0332			9.14	
						70										11.76	.50	[illegible]	25.88	1667	.2346	.1572			18.30	
						80													23.35	1504	.1620	.1134			16.37	
						90													20.48	1319	.1123	.0755			14.48	
						100													17.27	1112	.0766	.0535			12.20	
						110													13.32	864	.0503	.0352			9.69	
						70										17.64	.50	2.42	27.23	1754	.2467	.1727			19.25	
						80													24.70	1591	.1714	.1200			17.49	
						90													21.83	1406	.1197	.0838			15.44	
						100													18.62	1197	.0826	.0578			13.16	
						110													15.07	991	.0553	.0387			10.66	
						10	3.21	.05	6.73	11.77	12.08	.12	33.08	1.00	1.00	0	0	0	32.91	2119	14.52	10.16	1.18	1.01	23.25	REVERSE OPERATION
						20	[illegible]	.21	[illegible]	23.4?	24.08	.46							32.41	2087	3.60	2.52			22.92	
						30	[illegible]	.48	[illegible]	35.17	36.13	1.04							31.96	2032	1.56	1.09			22.32	
						40	12.??	.65	12.3?	46.72	48.17	1.65							30.36	[illegible]	.14	.59			21.47	

[illegible]

CALCULATION NOTES

• REQUIRED PUMP HEAD RISE

V_0 = CRAFT SPEED

Q = FLOW RATE ~ FT^3/SEC

$H_{P_{REQ}} = .00339\ Q^2 - .0078\ V_0^2$ (DERIVATION # C)

• ESTIMATED THRUST (POWER LIMIT)

V_0 = CRAFT SPEED

SHP = PUMP INPUT POWER

Q_{EQ} = EQUILIBRIUM FLOW RATE @ $H_{P_{AVAIL}} = H_{P_{REQ}}$.

A_j = EXIT AREA = 2.404 FT^2

V_j = EXIT VELOCITY = Q_{EQ}/A_j

T = THRUST = $\rho Q (V_j - V_0)$

P.C. = PROPULSIVE COEF. = $\dfrac{T V_0}{550\ SHP}$

• ESTIMATED THRUST (CAVITATION LIMIT)

V_0 = CRAFT SPEED

Q_{EQ} = EQUILIBRIUM FLOW RATE @ $H_{P_{CAV}} = H_{P_{REQ}}$

A_j = EXIT AREA = 2.404 FT^2

V_j = EXIT VELOCITY = Q_{EQ}/A_j

T = THRUST = $\rho Q (V_j - V_0)$

$H_{P_{EQ}}$ = EQUILIBRIUM HEAD RISE @ $H_{P_{CAV}} = H_{P_{REQ}}$

η_p = PUMP EFFICIENCY = .76 (POWER LIMIT CALC.)

SHP = PUMP INPUT POWER = $\dfrac{\rho g\ H_{P_{EQ}}\ Q_{EQ}}{550\ \eta_p}$

P.C. = PROPULSIVE COEF. = $\dfrac{T V_0}{550\ SHP}$

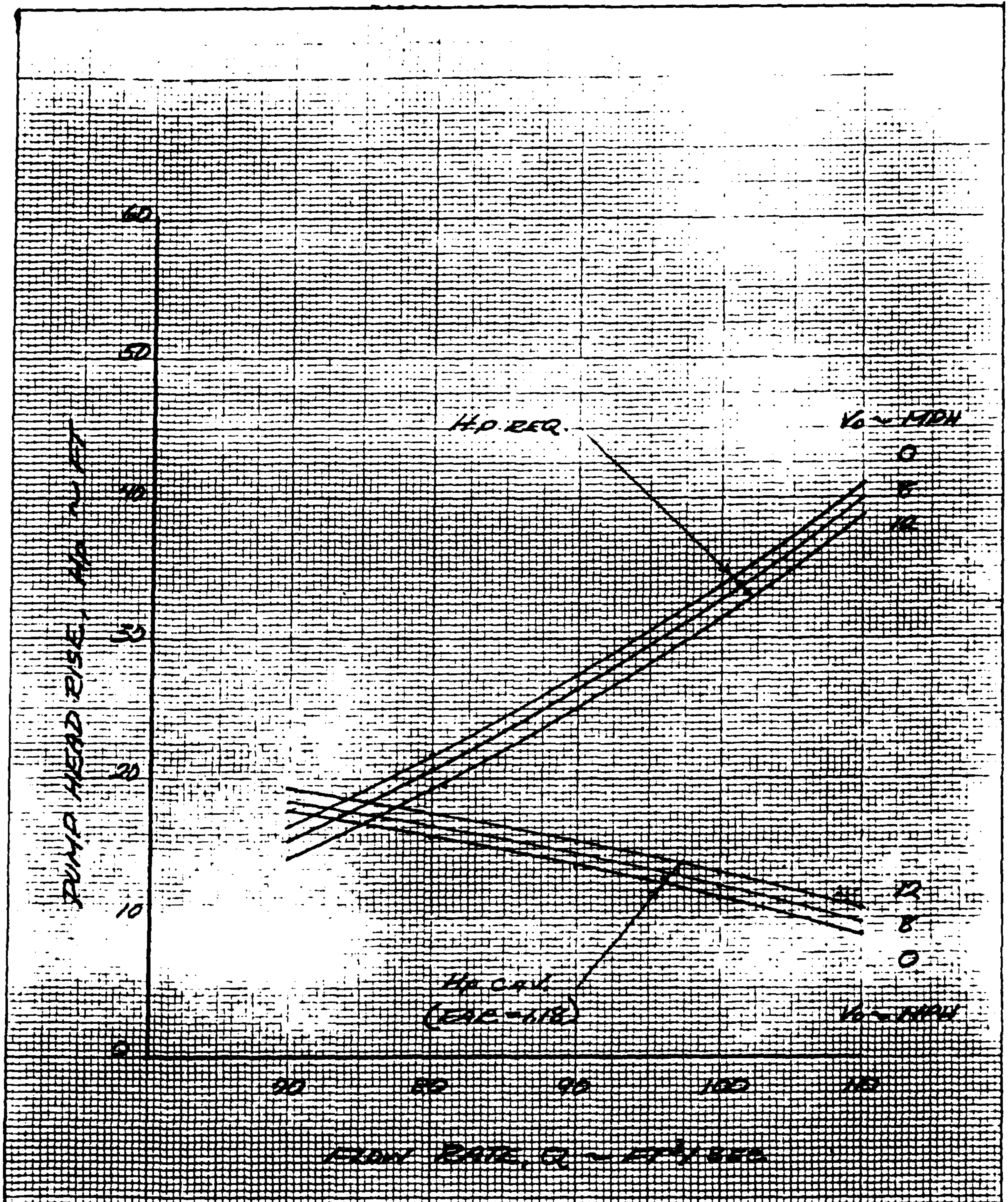
[illegible]

CALCULATIONS

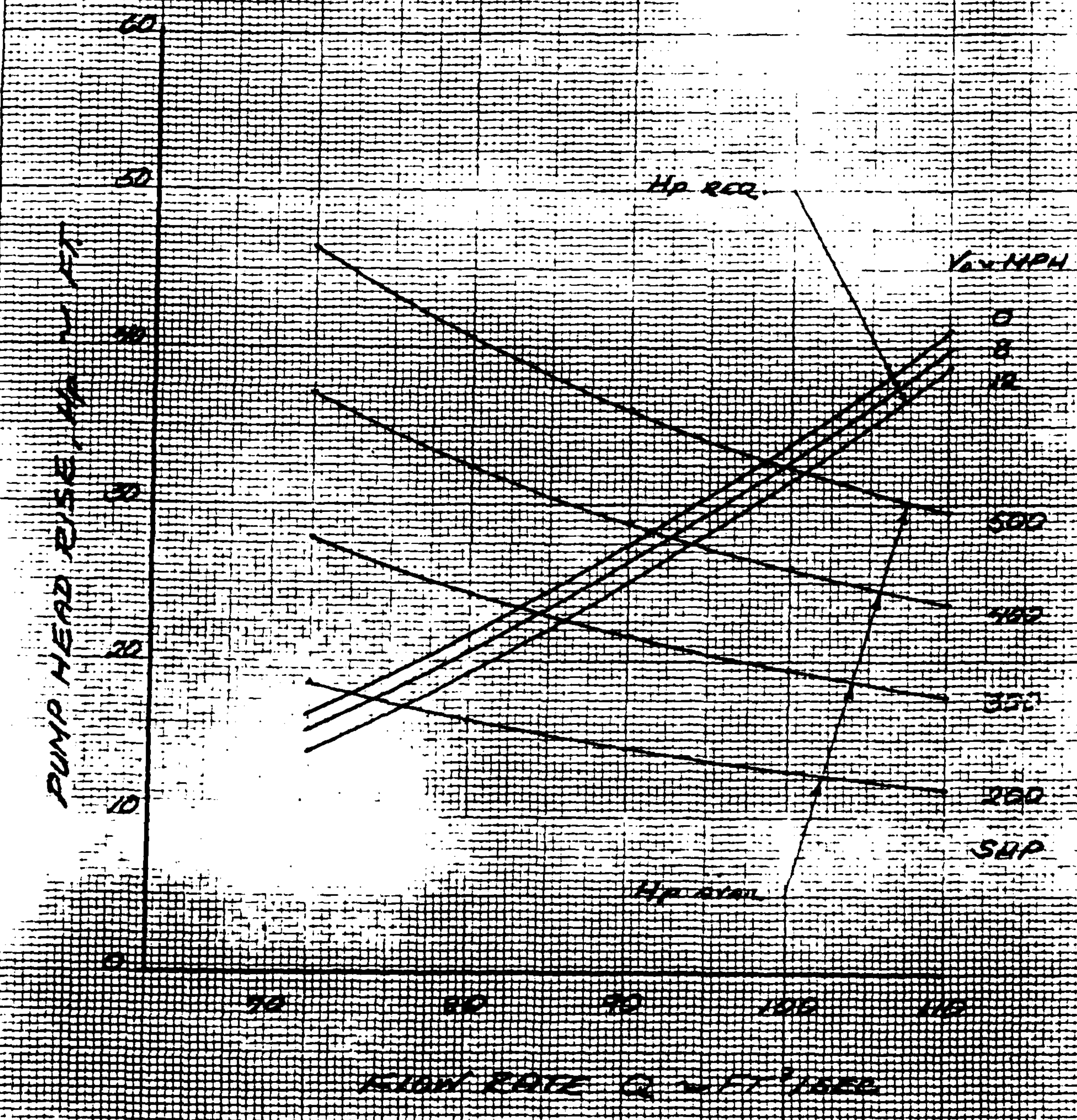
• REQUIRED PUMP HEAD FOR [illegible]

V_0	Q	H_{req}
0	70	16.61
	80	21.70
	90	27.46
	100	33.90
	110	41.02
5.88	70	16.34
	80	21.43
	90	27.19
	100	33.63
	110	40.75
11.76	70	15.53
	80	20.62
	90	26.38
	100	32.82
	110	39.94
17.64	70	14.18
	80	19.27
	90	25.03
	100	31.47
	110	38.59

NOT PLOTTED (CLARITY)



CAVITATION LIMITS



POWER LIMITS

Calculations (Cont.)

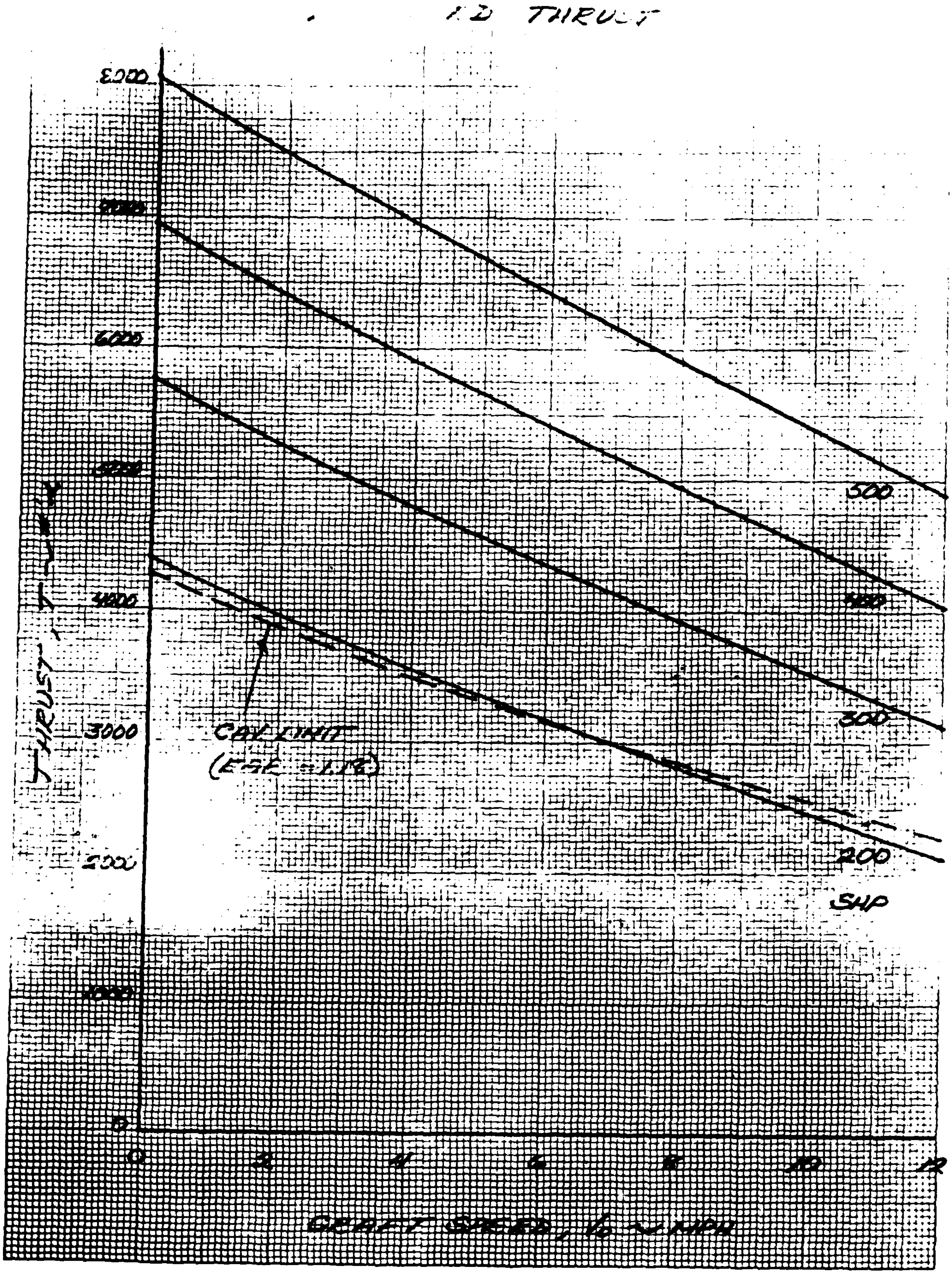
- Estimated Thrust (Power Limit)

V_o	SHP	Q_{EQ}	A_J	V_J	T	P.C.
0	500	98.5	2.404	40.97	8071	0
	400	91.4		38.02	6950	0
	300	83.2		34.61	5759	0
	200	72.7		30.24	4397	0
11.76	500	99.6		41.43	5910	.2527
8.MPH	400	92.6		38.52	4956	.2649
	300	84.4		35.11	3941	.2809
	200	74.0		30.78	2815	.3009
17.64	500	100.8		41.93	4897	.3141
12MPH	400	94.0		39.10	4034	.3235
	300	86.0		35.77	3118	.3333
	200	76.0		31.61	2123	.3405

CALCULATIONS (CONT.)

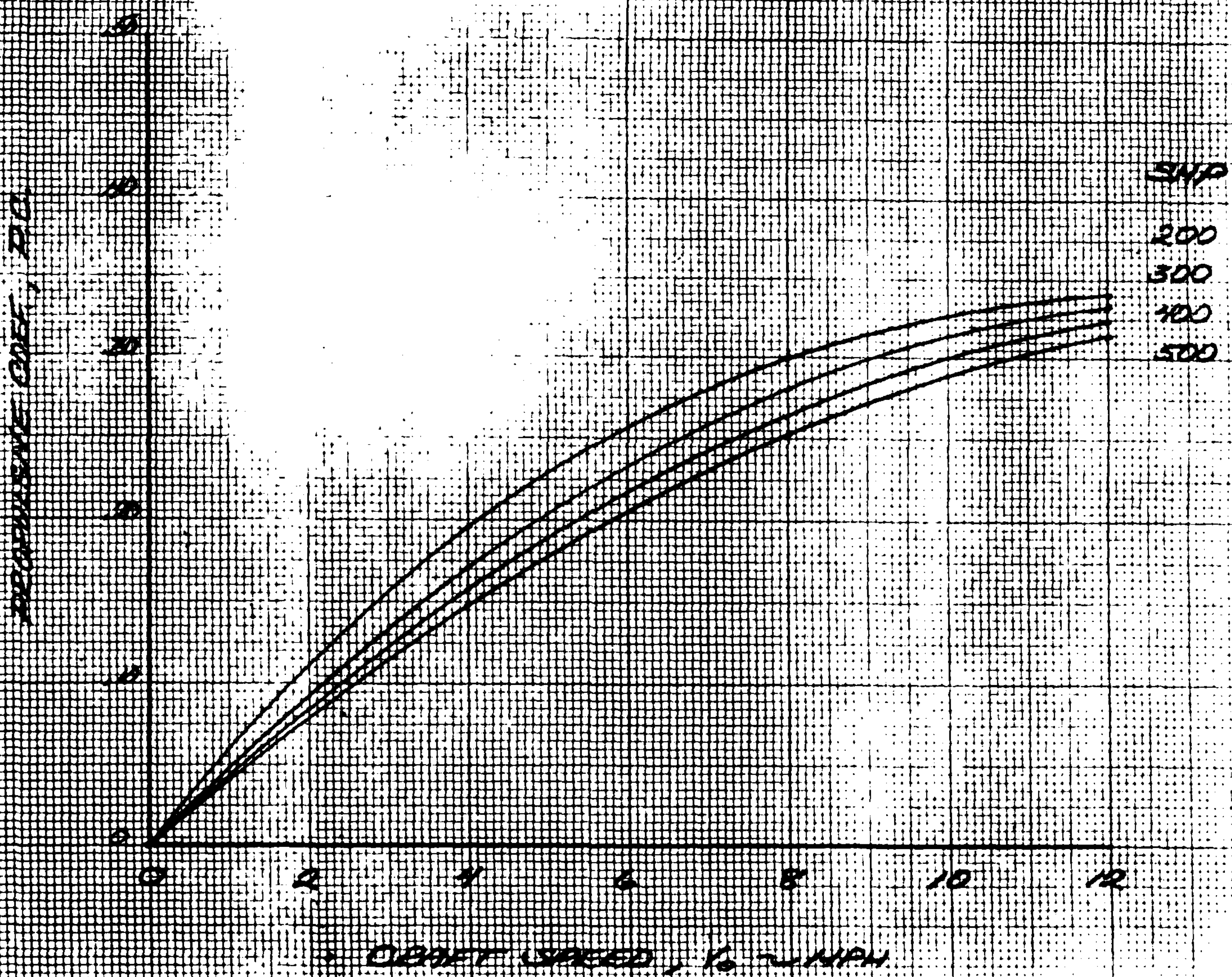
- ESTIMATED THRUST (CAVITATION LIMIT)

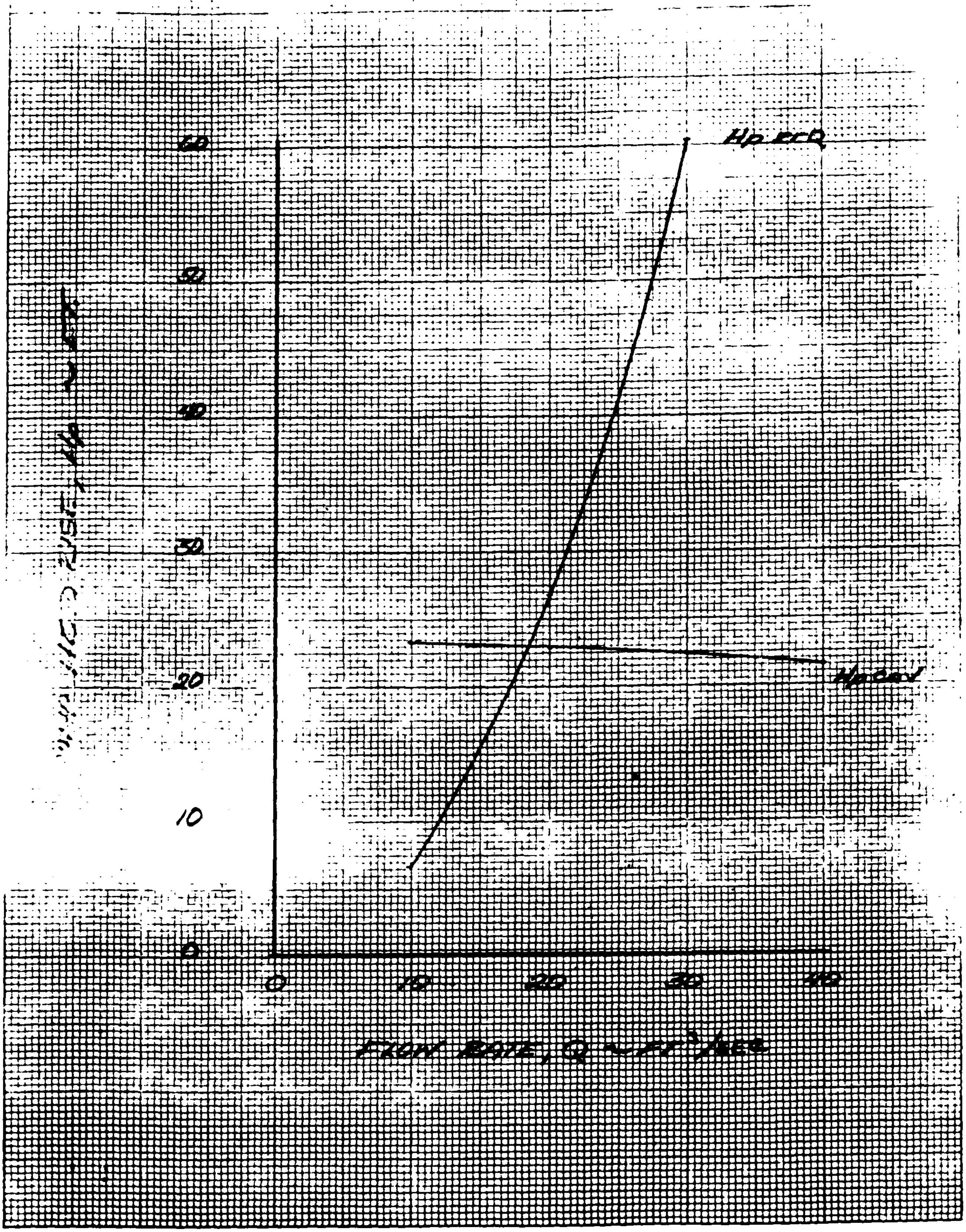
V_0	Q_{eq}	A_J	V_J	T	H_{req}	η_P	SHP	P.C.
0	71.7	2.404	29.43	4278	17.4	.76	192	0
11.76	74.3		30.91	2846	17.6		201	.3027
17.64	77.6		32.28	2272	18.0		215	~~.3389~~



22-116 · CROSS SECTION 20 SQUARES TO INCH

ESTIMATED PROPULSIVE COEFFICIENT





Hp req
Hp avl
10
20
30
40
0
FLOW RATE, Q ~ FT³/SEC

REQUIRED PUMP HEAD

$H_{PREQ} = H_{LR} + H_{LI} - H_O$

$$H_{LR} = .0666\,Q^2$$
$$H_{LI} = .000533\,Q^2$$
DERIVATION #1

$$H_O = 0$$

$$H_{PREQ} = .0666Q^2 + .000533Q^2 - 0 = .0671\,Q^2$$

Q	H_{PREQ}
10	6.71
20	26.84
30	60.39
40	107.36

CAVITATION LIMIT (SEE CAV. LIMIT CALCS.)

Q	H_{PAV}
[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	21.72
[illegible]	[illegible]

[illegible]

Q_{REQ}	A_R	V_R	T_R	H_{PREQ}	η_P	SHP
18.5	.764	24.21	896	23.1	.76	66

[illegible]

REF. [illegible]

VAN LAMEREN FIG. 1
" FIG. 2 } MODEL TESTS OF PROP. IN AXIAL CYLINDER

DERIVATIONS

#1 ESTIMATED INLET, [illegible] & REVERSE SYSTEM LOSSES

#2 REQUIRED PUMP HEAD [illegible]

#3 PROPELLER BLADE AREA

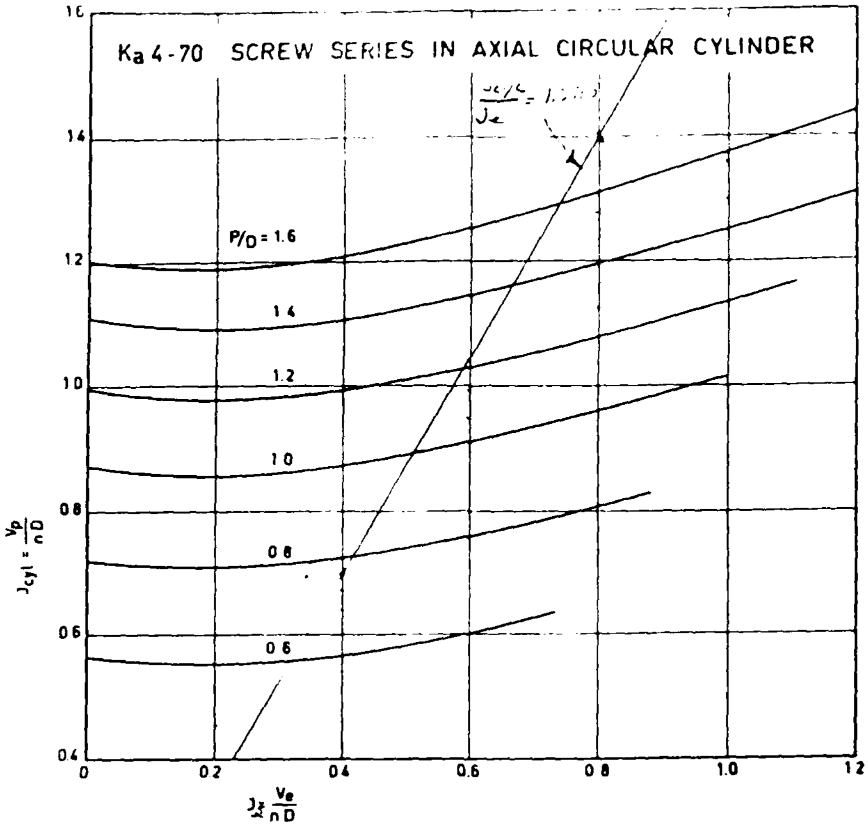


Fig. 29 Relation between velocity of "screw + cylinder" combination and velocity in cylinder

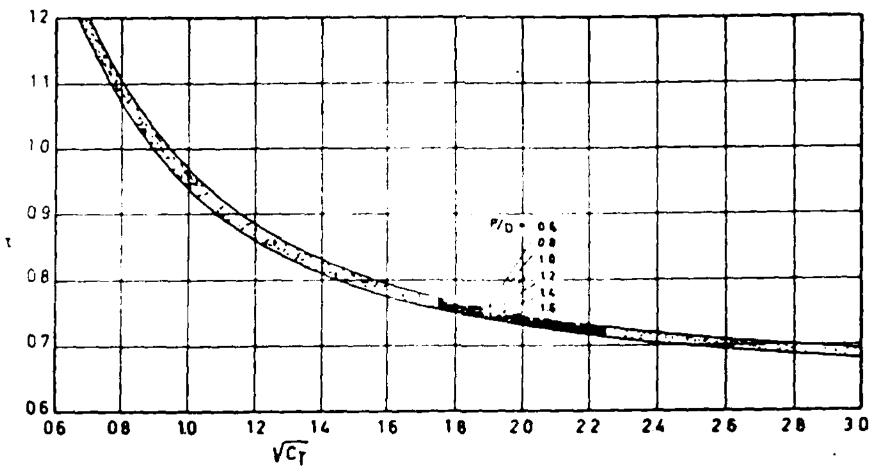


Fig. 30 Relation between thrust coefficient C_T and thrust ratio τ of nozzle no. 19a

figure have been obtained by substituting nozzles with different length-diameter ratios by systems of annular vortexes and calculating the induced velocities in the screw disk.

If the radial displacement of the streamlines is small, we can consider the streamlines as lying approximately on cylindrical planes. If internal friction and turbulence are neglected, the radial

FIG. 1

VON LAMEREN

A-20

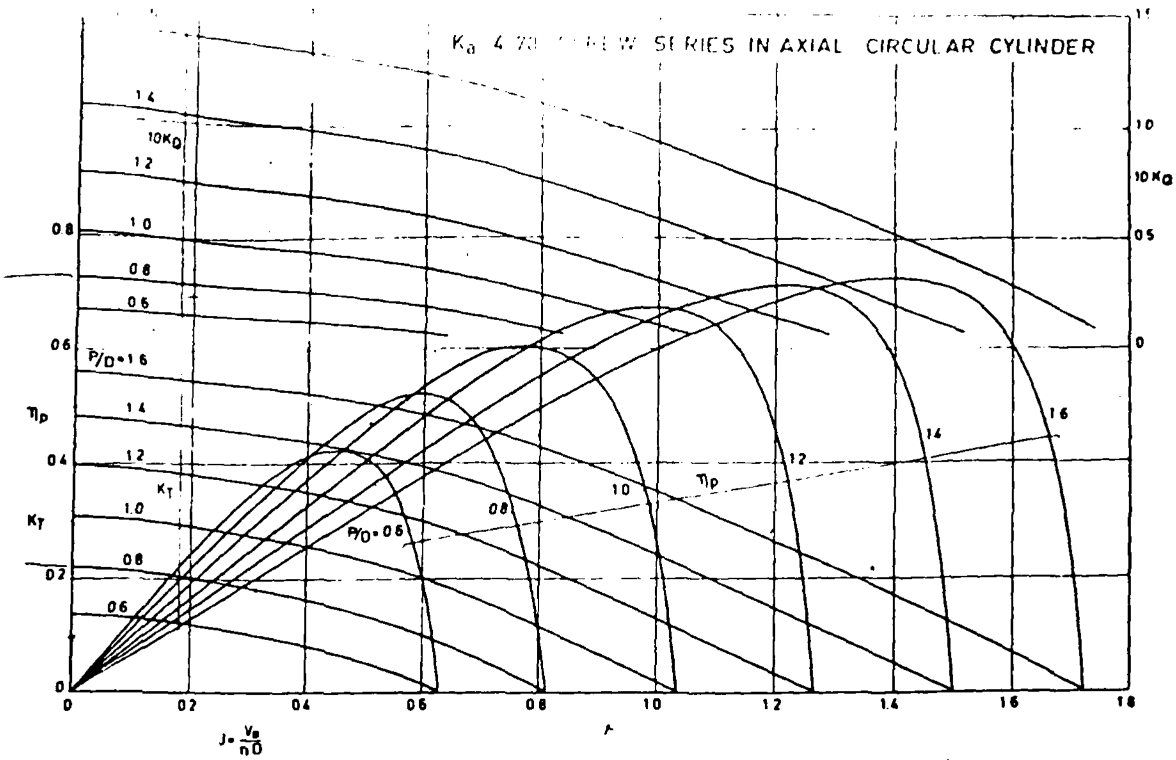


FIG. 2

Fig. 28 Results of open-water tests with Ka 4-70 screw series in an axial cylinder

been obtained from the experiments with the Ka 4-70 screw series in an axial circular cylinder and from the application of the momentum theorem

From the comparison of the axial velocities obtained with these methods, we see that

1 The velocities agree reasonably well at high loadings of the ducted propeller system ($C_T > 1$)

2 The difference between the axial velocities becomes very large at low loadings ($C_T < 1$).

In regard to the second conclusion, the following remark may be made. From Fig. 13 it can be seen that the nozzle drag due to friction becomes substantial at low loadings of the ducted-propeller system. Then, it is no longer permitted to neglect the effect of friction on the force action between nozzle and fluid.

The design of a screw in a nozzle may now be carried out as follows:

With given thrust T or power P, intake velocity V_e, and number of revolutions n, the B_p and consequently the optimum diameter coefficient D can be determined with the aid of open-water test results of the nozzle considered, in combination with a systematic screw series (see, for instance, Fig. 24). In addition, the thrust coefficient C_T and the propeller thrust-total thrust ratio τ can be determined. With the aid of the experiments of the systematic screw series in the axial circular cylinder or using the momentum theorem, the axial velocity V_P in the way of the screw can be found. In addition, the mean axial velocity in the vicinity of the screw due to the nozzle action, U_N, and due to the screw action U_P, can be calculated.

The pressure difference created by the screw becomes

$$\Delta p = \frac{T_p}{\frac{\pi}{4}(D^2 - d_h^2)}$$

In order to avoid an excessive loading of the inner radii of the screw blades, the usual assumption for axial pumps that the head is constant for all radii is abandoned. The following radial $\Delta p(r/R)$ distribution is suggested for the screws in nozzle no. 19a:

$$\Delta p(r/R) = [4.88 - 4r/R]\cdot[r/R - 0.133]\Delta p$$

The radial distribution of the axial and tangential velocities at the screw may be approximated as follows:

A reasonable radial distribution of the axial velocities due to the nozzle action can be determined from Fig. 32. The results given in this

Von Lamgren

A-21

INLET ENTRANCE

A_e = ENTRANCE AREA $(6 \times 2)(1)$ = 8.00 FT^2

Q = NOMINAL FLOW RATE = 25 FT^3/SEC

V_e = ENTRANCE VELOCITY = Q/A_e = 25.00/8.00 = 9.325 FT/SEC

K_e = LOSS COEF. = 1.50 (CRANE - SQ. EDGED ORIFICE)

$$HL_e = K_e \frac{V_e^2}{2g} = \frac{(1.50)(9.325)^2}{2(32.2)} = 2.047'$$

INLET FRICTION & BEND

A_I = INLET AREA $= \frac{(22)(24)}{144}$ = 3.667 FT^2

V_I = INLET VELOCITY = Q/A_i = 25/3.667 = 20.45 FT/SEC

$$d_e = \text{EQUIV. INLET DIAM.} = \frac{4A_i}{2\left(\frac{22+24}{12}\right)} = \frac{4(3.667)}{2\left(\frac{22+24}{12}\right)} = 1.9132'$$

$$R_e = \frac{V_i d_e}{1.24\times10^{-5}} = \frac{(20.45)(1.9132)}{1.24\times10^{-5}} = 3.16\times10^6$$

e/d_e = RELATIVE [illegible] = .000005/1.9132 = .00000261

f = FRICTION F[illegible] = .00925

L_i = INT[illegible] = 3'

$$L_E = [illegible] \cdot \left(\frac{L}{D}\right)_E (n)\left(\frac{\alpha}{90}\right) = (36)(1.9132)\left(\frac{25}{90}\right) = 19.13'$$

L = TOTAL EQUIV. [illegible] = $L_i + L_E$ = 3 + 19.13 = 22.13'

$$HL = f\left(\frac{L}{d_e}\right)\left(\frac{V_I^2}{2g}\right) = (.00925)\left(\frac{22.13}{1.9132}\right)\frac{(20.45)^2}{2(32.2)} = .732'$$

[illegible]

SHAFT

C_D = DRAG COEF. DUE TO CROSS FLOW = $1.1 \sin^3 \frac{\alpha}{2} = 1.1 \sin^3 \frac{25}{2}$ = .0112

V_I = 20.45 FT/SEC

ℓ = SHAFT LENGTH = 4'

d = SHAFT DIAM = [illegible]" = .1667'

D = SHAFT DRAG = $C_D \frac{\rho}{2} V_I^2 \ell d = (.0112)(\frac{3}{2})(20.45)^2(4)(.1667)$ = 3.12#

$HL = \frac{D}{\rho g A_I} = \frac{3.12}{2(32.2)(3.667)}$ = .0132'

TRANSITION

A_T = CROSS SECTION AREA = $\frac{5.667 + [illegible]}{2}$ = 2.8806 FT²

V_T = VELOCITY = Q/A_T = 75/2.8806 = 26.04 FT/SEC

d_E = EQUIV. DIAM. = $\sqrt{[illegible]}$ = [illegible]'

$Re = \frac{V_T d_E}{1.24 \times 10^{-5}} = \frac{(26.04)([illegible])}{1.[illegible]}$ = 4.02 × 10⁶

ℓ/d_E = RELATIVE [illegible] = .00000261

f = FRICTION FACTOR = .0094

L_T = [illegible] = .5[illegible]'

$HL = f\left(\frac{L}{d_E}\right)\frac{V_T^2}{2g}$ = [illegible] $\left(\frac{.5[illegible]}{1.9156}\right)\frac{(26.[illegible])^2}{2(32.2)}$ = .0301'

Bracing Tube

A_p = CROSS SECTION AREA = 2.0942 ft^2

V_p = VELOCITY = Q/A_p = [illegible] / 2.0942 = 35.81 ft/sec

ℓ = TUBE LENGTH = 2'

$$R_e = \frac{V_p \ell}{1.24 \times 10^{-5}} = \frac{(35.81)(2)}{1.24 \times 10^{-5}} = 5.78 \times 10^6$$

C_f = .00321

S = TUBE WETTED SURF. = (2)(.33) π = 2.07 ft^2

D = TUBE DRAG = $(C_f + .00[illegible])(S)(\rho/2) V_p^2 = (.0032 + .000[illegible])(2.07)(\rho/2)(35.81)^2$ = 10.64#

$$HL = \frac{D}{\rho g A_p} = \frac{10.64}{(2)(32.2)(2.09[illegible])} = .0789'$$

Struts

A_p = CROSS SECTION [illegible] ft^2

V_p = VELOCITY = Q/A_p [illegible]

c = STRUT [illegible]

R_e = $\frac{V_p c}{[illegible]}$ [illegible] = 9.62 $\times 10^5$

C_f = .00442

t/c = [illegible] .0958

$C_D = 2(C_f + .0008)(1 + 1.2\, t/c)$ = 2(.00442 + .0008)(1 + 1.2 × .0958) = .0116

S = STRUT PLANFORM AREA = 2(1.66[illegible] - .[illegible])(.333) = .8884 ft^2

D = STRUT DRAG = $C_D S \frac{\rho}{2} V_p^2 = (.0116)(.8884)(\rho/2)(35.81)^2$ = 13.22#

$$HL = \frac{D}{\rho g A_p} = \frac{13.22}{2(32.2)(2.09[illegible])} = .0980'$$

[illegible] INLET LOSS

INLET ENTRANCE	2.0400
INLET FRICTION & [illegible]	.0320
[illegible]	.0132
TRANSITION	.0301
[illegible] TUBE	.0289
[illegible]	.0980

$$HL_I = 2.9992$$

$$k = \frac{HL_I}{Q^2} = \frac{2.9992}{(75)^2} = .000533$$

$$HL_I = .000533\,Q^2$$

$\underline{5}$

Casing Friction

Q = NOMINAL FLOW RATE = [illegible]

A_p = [illegible] AREA = [illegible] FT²

V_p = [illegible] Q/A_p [illegible] 35.81 FT/SEC

d = CASING DIAM. = 20" [illegible]'

$Re = \frac{V_p d}{\nu}$ = [illegible] $= 4.81 \times 10^{6}$

e/d = RELATIVE ROUGHNESS = $.000005/1.667 = .000003$

f = FRICTION FACTOR = .0092

L_c = CASING LENGTH = 4'

$HL = f\left(\frac{L_c}{d}\right)\frac{V_p^2}{2g} = (.0092)\left(\frac{4}{1.667}\right)\frac{(35.81)^2}{2(32.2)} = .4396'$

Casing Divergence

A_p = CROSS SECT. AREA OF [illegible] = 2.0942 FT²

A_J = CROSS SECT AREA OF [illegible] $= \frac{(20)(18)-(8)^2+(285)[illegible]^2}{144} = 2.404$ FT²

K = LOSS COEF. $= \left(1-\frac{A_p}{A_J}\right)^2 = .0166$ (CRANE - SUDDEN EXP.)

V_p = [illegible] Q/A_p [illegible] 35.81 FT/SEC

$HL = K\frac{V_p^2}{2g}$ = [illegible] = .3305'

[illegible]

[illegible]

A_R = CROSS SECT. AREA [illegible] 2.404 FT^2

V_R = VELOCITY = Q/A_R = 75/2.404 = 31.20 FT/SEC

c = [illegible] CHORD = 26.25/12 = 2.19'

$R_e = \frac{V_R c}{1.24\times10^{-5}} = \frac{(31.20)(2.19)}{1.24\times10^{-5}}$ = [illegible] $\times 10^6$

C_f = .00323

t/c = RUDDER THICKNESS RATIO = .75/26.25 = .03

$C_D = 2(C_f + .0008)(1+1.2\,t/c) = 2(.00323+.0008)(1+1.2\times.03)$ = .0084

S = RUDDER PLANFORM AREA = (2.19)(1.667) = 3.66 FT^2

D = RUDDER DRAG = $C_D S \frac{\rho}{2} V_R^2 = (.0084)(3.66)(2/2)(31.20)^2$ = 29.93#

$HL = \frac{D}{\rho g A_R} = \frac{29.93}{2(32.2)(2.404)}$ = .193'

Total Casing Loss

CASING FRICTION	[illegible]
CASING DIVERGENCE	[illegible]
RUDDER	[illegible]

HL_c = [illegible]'

$k = \frac{HL_c}{Q_{NOM}^2}$ = [illegible] .000171

HL_c = .000171 Q^2

REVERSE ELBOW

K = LOSS COEF. = $1 + 1.5 = 2.5$ (STD + SQ EDGE ... - ...)

A_R = REVERSE JET AREA = $\frac{(20)(5.5)}{144}$ = .764 FT^2

V_R = REVERSE JET VELOCITY = Q/A_R = 75/.764 = 98.17 FT/SEC

$HL = K \frac{V_R^2}{2g} = (2.5)\frac{(98.17)^2}{2(32.2)} = 374.12'$

TOTAL REVERSE SYSTEM LOSS

CASING FRICTION	.44
CASING DIVERGENCE	.33
REVERSE ELBOW	374.12

$HL_R = 374.89'$

$k = \frac{HL_R}{Q_{NOM}^2} = \frac{374.89}{(75)^2} = .0666$

$HL_R = .0666\ Q^2$

REQUIRE[illegible]

$H_{P\,REQ} = HL_J + HL_I + HL_E - H_O$ [illegible]

$HL_J = $ [illegible]

$V_J = Q/A_J$

$A_J = 2.404\ FT^2$

$$HL_J = \frac{Q^2}{(2.404)^2(2)(32.2)} = .00269\ Q^2$$

$HL_I = .000555\ Q^2$

$HL_E = .000151\ Q^2$

DERIVATION #1

$$H_O = (RPC)\frac{V_O^2}{2g}$$

$RPC = .50$

$$H_O = \frac{(.50)\ V_O^2}{2(32.2)} = .0078\ V_O^2$$

$$H_{P\,REQ} = .00269\,Q^2 + .000555\,Q^2 + .000151\,Q^2 - .0078\,V_O^2$$

$$= .00339\,Q^2 - .0078\,V_O^2$$

[illegible]

[illegible]

r	c	T.M.	f(c)
2	10.70	1/2	5.35
3	12.25	1	12.25
4	13.70	1	13.70
5	14.94	1	14.94
6	16.00	1	16.00
7	16.90	1	16.90
8	17.50	1	17.50
9	17.85	1	17.85
10	[illegible]	1/2	9.00
			123.49

$A_x = (3)(1)(123.49) = 370.47\ in^2$

$A_D = (.785)(20)^2 = 314\ in^2$

$E.A.R. = \frac{370.47}{314} = 1.18$

PROJECTED AREA

r	c	α	$c_{\cos α}$	T.M.	f(c)
2	10.70	[illegible]	5.69	1/2	2.84
3	12.25	[illegible]	8.40	1	8.40
4	13.70	38.51	10.72	1	[illegible]
5	[illegible]	[illegible]	12.60	1	[illegible]
6	[illegible]	27.95	14.13	1	[illegible]
7	[illegible]	[illegible]	15.39	1	[illegible]
8	[illegible]	21.70	16.26	1	[illegible]
9	[illegible]	[illegible]	16.83	[illegible]	[illegible]
10	[illegible]	[illegible]	17.15	[illegible]	[illegible]
					[illegible]

$\alpha = \text{ARC TAN}\left(\frac{P}{2\pi}\right)/r$

$= \text{ARC TAN}\left(\frac{20}{2\pi}\right)/r$

$= \text{ARC TAN } 3.1831/r$

$A_p = (3)(1)(105.74) = 317.22\ in^2$

$A_0 = 314\ in^2$

$PAR = \frac{317.22}{314} = 1.01$

STRUCTURAL [illegible]

- PROPELLER
- SHAFT
- RUDDER
- DUCT

PROPELLER [illegible]

R = PROP. RADIUS = 10"

Z = NO. OF BLADES = 3

P = PITCH = 20"

T = PROP. THRUST = $\rho g H_p A_i$ = (64.4)([illegible])(2.00[illegible]) = 3115#

N = PROP. SPEED = $\frac{60\,Q}{A_p V_{eye} D} = \frac{(60)(18.5)}{(2.07[illegible])([illegible])(1.66)}$ = 355 RPM

Q' = PROP. TORQUE = $63024 \frac{SHP}{N} = \frac{(63024)(66)}{355}$ = 11717 IN#

REVERSE

$a = \frac{2\pi R}{P} = \frac{2\pi(10)}{20} = 3.1416$

$x = r/R$

$K = f(x)$ TABLE I CONOLLY

$A_1 = f(a,x)$ " II "

$A_2 = f(a,x)$ " III "

$B_1 = f(a,x)$ " [illegible] "

$B_2 = f(a,x)$ " [illegible] "

$C_1 = f(x)$ " [illegible] "

c = SECTION CHORD

t = MAX. SECTION THICKNESS

σ_R = SPANWISE BENDING STRESS $= \frac{RK}{Zct^2}\left[A_1\left(\frac{2\pi RT}{P}\right) + A_2(\ldots)\right]$

σ_θ = CHORDWISE BENDING STRESS $= \frac{RK}{Zct^2}\left[B_1\left(\frac{2\pi RT}{P}\right) + B_2\left(\frac{Q}{R}\right)\right]$

σ_c = CENTRIFUGAL STRESS $= \frac{2240\,N^2R^2C_1}{10^{10}}$

$s_{T_{MAX}}$ = MAX. TENSILE STRESS (FACE) $= \sigma_R + \sigma_c$

$s_{S_{MAX}}$ = MAX. SHEAR STRESS (FACE) $= \frac{s_T - \sigma_\theta}{2}$

PROPELLER STRESS CALCULATION

BENDING STRESSES

R	Z	P	T	Q'	a	x	K	A_1	A_2	c	t	σ_R	B_1	B_2	σ_θ
10.00	3	20	3115	11717	3.1416	.20	.1203	6.90	57.95	10.70	.725	9586	2.26	19.30	3169
						.30	.1207	6.94	39.26	12.75	.638	9192	3.13	18.69	4238
						.40	.1007	7.27	31.20	13.70	.550	8723	3.97	19.25	[illegible]
						.50	.0720	7.46	27.63	14.94	.463	8206	4.79	19.98	5267
						.60	.0437	8.14	27.00	16.00	.375	7585	5.63	20.33	5155
						.70	.0214	10.00	27.95	16.90	.288	6647	6.58	[illegible]	4601
						.80	.0075	11.61	30.10	17.50	.201	5265	7.65	[illegible]	3633
						.90	.0013	12.53	30.32	17.65	.113	3007	8.44	24.74	2121

CENTRIFUGAL STRESS

R	N	x	C_1	
10.00	355	.20	16.6	47
		.30	12.0	34
		.40	9.5	27
		.50	7.7	22
		.60	6.2	18
		.70	4.8	14
		.80	3.4	10
		.90	1.9	5

COMBINED STRESSES

x	σ_R	σ_θ	σ_c	S_{TMAX}	[illegible]
.20	9586	3169	47	9633	3222
.30	9192	4238	34	9226	2494
.40	8723	4974	27	8750	1888
.50	8206	5267	22	8230	1482
.60	7585	5155	18	7603	1224
.70	6647	4601	14	6661	1030
[illegible]	5265	3633	10	5275	821
[illegible]	3007	2121	5	3012	446

[illegible] SHAFT

TORSIONAL STRESS

Q = TORSION MOM. IN SHAFT = 11717 IN#

d = SHAFT DIAM. = 2" , r = 1"

J = POLAR MOM. OF INERTIA = $\frac{\pi}{2}(1)^4$ = 1.5708

s_s = TORSIONAL STRESS IN SHAFT = $\frac{Qr}{J} = \frac{(11717)(1)}{1.5708}$ = 7459 psi

FACTOR OF SAFETY = $\frac{20000}{7459}$ = 2.68 ON SHEAR YIELD (6061-T6)

WHIRLING FREQUENCY

w = WEIGHT PER UNIT LENGTH OF SHAFT = $(.785)(2)^2(1)(.096)$ = .3014 #/in

l = DISTANCE BETWEEN SUPPORTS = 82"

I = MOM. OF INERTIA OF SHAFT = $.049\,d^4 = (.049)(2)^4$ = .784 in^4

D = STATIC DEFLECTION OF SHAFT DUE TO OWN WEIGHT

$$= .00542\,\frac{wl^4}{EI} = \frac{(.00542)(.3014)(82)^4}{(10{,}200{,}000)(.784)} = .0092''$$ (FREE - FIXED)

f = WHIRLING FREQUENCY = $\frac{3.55}{D^{1/2}} = \frac{3.55}{(.0092)^{1/2}}$ = 36.[illegible] cps

= 2216 RPM

$$N_{DES} = \frac{(74.3)(60)}{(2.0412)(.875)(1.667)}$$ = 14[illegible] (EMPIRICAL [illegible])

STRUCTURE

PLATE

p = DESIGN PRESSURE = 23.1' = 1[illegible] (REVISE [illegible])

ℓ = SPAN = 19.5"

b = PANEL WIDTH = 26"

M = BENDING MOMENT IN PLATE = $\frac{p\ell^2 b}{8} = \frac{(10.3)(19.5)^2(26)}{8}$ = 12803 IN#

t = PLATE THICKNESS = .75"

Z = SECTION MODULUS OF PLATE = $\frac{bt^2}{6} = \frac{26(.75)^2}{6}$ = 2.4375 IN^3

s_M = BENDING STRESS IN PLATE = $\frac{M}{Z} = \frac{12803}{2.4375}$ = 5253 psi

a = EFFECTIVE MOMENT ARM = 2.66"

Q = RUDDER TORQUE = $p\,\ell\, b\, a$ = (10.36)(20)(26)(2.66) = 14330 IN#

s_Q = SHEAR STRESS IN PLATE = $\frac{3Q}{bt^2} = \frac{3(14330)}{26(.75)^2}$ = 2939 psi

$s_{S_{MAX}}$ = MAX. COMBINED SHEAR STRESS IN PLATE = $\sqrt{\left(\frac{s_M}{2}\right)^2 + (s_Q)^2}$

$= \sqrt{\left(\frac{5253}{2}\right)^2 + (2939)^2}$

= 3942 psi

FACTOR OF SAFETY = $\frac{35000/2}{3942}$ = 4.44 ON YIELD (6061-T6)

= $\frac{40000/2}{3942}$ = 5.07 ON YIELD (POLY[illegible] G[illegible]

STOCK

Q = RUDDER TORQUE = 14330 IN#

d = STOCK DIAM. = 2.00", r = 1.00"

J = POLAR MOM. OF INERTIA = $\frac{\pi r^4}{2} = \frac{\pi(1)^4}{2}$ = 1.5708

s_s = SHEAR STRESS IN STOCK = $\frac{Qr}{J} = \frac{(14330)(1)}{1.5708}$ = 9123 psi

FACTOR OF SAFETY = $\frac{30000}{9123}$ = 3.29 ON WELDED YIELD (316 STAINLESS)

[illegible] [illegible]

FLANGE:

Q = TORSION IN RUDDER STOCK = 14330 IN#

d = STOCK DIAM = 2.00" r = 1.00"

t = FLANGE THICKNESS = .25"

A_s = SHEAR AREA IN FLANGE = $\pi d t = \pi(2)(.25) = 1.5708\ IN^2$

s_s = SHEAR STRESS IN FLANGE = $\frac{Q}{A_s r} = \frac{14330}{(1.5708)(1)}$ = 9123 psi

FACTOR OF SAFETY = $\frac{30{,}000}{9123}$ = 3.29 ON WELDED YIELD (316 STAINLESS)

ATTACH BOLTS

p = DESIGN PRESSURE = 10.36 psi

S = RUDDER AREA = (20)(26) = 520 IN^2

F = LOAD NORMAL TO RUDDER: $pS = (10.36)(520) = 5387^{\#}$

A_T = TENSILE AREA IN BOLTS = $6(.047) = .2820\ IN^2$ 12, 5/16-18 BOLTS

s_T = TENSILE STRESS IN BOLTS = $\frac{F}{A_T} = \frac{5387}{.2820}$ = 19103 psi

FACTOR OF SAFETY = $\frac{30000}{19103}$ = 1.57 ON YIELD (316 STAINLESS)

PLATE

p = DESIGN PRESSURE = 23.1' = 10.36 #/"

l = SPAN = 20"

w = UNIT WIDTH = 1"

M = BENDING MOMENT IN PLATE = $\frac{p l^2 w}{12} = \frac{(10.36)(20)^2(1)}{12} =$ 345.33 IN #/IN WIDTH

t = PLATE THICKNESS = .375"

Z = SECTION MODULUS OF PLATE = $\frac{w t^2}{6} = \frac{(1)(.375)^2}{6} =$ .0234 IN³/IN WIDTH

s = BENDING STRESS IN PLATE = $\frac{M}{Z} = \frac{345.33}{.0234} =$ 14758 psi

FACTOR OF SAFETY = $\frac{19000}{14758}$ = 1.29 ON WELDED YIELD (5086-H116)

$= \frac{40{,}000}{14758}$ = 2.71 ON FLEX. YIELD POLYESTER/GLASSCLOTH

Vertical Louvres

p = DESIGN PRESSURE = 10.36 psi

l = SPAN = 5"

c = CHORD = 5"

M = BENDING MOMENT = $\frac{p l^2 c}{12} = \frac{(10.36)(5)^2(5)}{12} = 107.92$ IN#

t = PLATE THICKNESS = .1875"

Z = SECTION MODULUS = $\frac{w t^2}{6} = \frac{(5)(.1875)^2}{6} = .0293$ IN³

s = BENDING STRESS = $\frac{M}{Z} = \frac{107.92}{.0293} = 3683$ psi

FACTOR OF SAFETY = $\frac{19000}{3683}$ = 5.16 ON WELDED YIELD (5086-H116)

Horizontal Louvres

p = DESIGN PRESSURE = 10.36 psi

l = SPAN = 13"

b = PANEL WIDTH = 5"

k = OVERLAP FACTOR = [illegible] = 1.23

M = BENDING MOMENT = $k \frac{p l^2 b}{8} = \frac{(1.23)(10.36)(13)^2(5)}{8} = 1346$ IN#

Z = SECTION MODULUS = $\frac{(.1875)(2)^2}{6} = .125$ IN³

s = BENDING STRESS = $\frac{M}{Z} = \frac{1346}{.125} = 10768$ psi

FACTOR OF SAFETY = $\frac{19000}{10768}$ = 1.76 ON WELDED YIELD (5086-H116)

E[illegible] CYLINDER (2.50" BORE [illegible])

M = REQUIRED MOMENT = 13588 IN#

A_{cyl} = CYL. AREA = $(.785)(2.50)^2 = 4.9063\ IN^2$

$p_{cyl\ avail}$ = CYL. PRESS. = 1000 psi

$F_{cyl\ avail}$ = AVAILABLE CYL. FORCE = $p_{cyl} A_{cyl} = (1000)(4.9063) = 4906$ #

r = CRANK RADIUS = 3.50"

Θ = TOTAL TRAVEL = 45°

a = EFFECTIVE MOMENT ARM = $r \cos \frac{\Theta}{2} = (3.50) \cos 22.5° = 3.23"$

$M_{avail} = \left(F_{cyl\ avail}\right)(a) = (4906)(3.23) = 15846$ (O.K.)

ℓ_{REQ} = REQ. STROKE = $2r \sin \frac{\Theta}{2} = 2(3.50) \sin 22.5 = 2.68"$ (O.K.)

ESTIMATED WEIGHTS

[illegible]

$A = (20)\pi(16)/144 = 6.9813\ FT^2$
$t = .375"$, $w = (.375)(144)(.096) = 5.18\ \#/FT^2$
$W = (6.9813)(5.18) = 36.16\ \#$

BACKING RING

$\nabla = .785\ [(22.75)^2 - (20.875)^2](1) = 64.21\ IN^3$
$w = .096\ \#/IN^3$
$W = 64.21(.096) = 6.16\ \#$

FLANGE

$\nabla = .785\ [(22.75)^2 - (20.75)^2](.5) = 34.15\ IN^3$
$w = .096\ \#/IN^3$
$W = (34.15)(.096) = 3.28\ \#$

[illegible]

$\nabla = 4(4)(.375)(.71)(\ldots) = 34.08\ IN^3$
$w = .096\ \#/IN^3$
$W = 34.08(.096) = 3.27\ \#$

BEARING TUBE

STA.	ℓ	[illegible]	T.M.	$f(\nabla)$
1	4.00	12.56	[illegible]	3.28
2	4.00	12.56	1	12.56
3	3.50	9.62	1	9.62
4	2.75	5.94	½	2.97
				31.43

$\nabla = 4(31.43) - (8)(.785)(2.625)^2 - (4)(.785)(2.75)^2 = 66.[illegible]$
$w = .096\ \#/IN^3$
$W = 6.39\ \#$

TOTALS

$W = 36.16$ $\times 1.50/2.66$
6.16
3.28
3.27
6.39
55.26 # CONV.

31.16 # PLAST.

CIRCULAR DUCT

$A = (20)\pi(3)/144 = 1.309\ \text{FT}^2$
$t = .375'' \quad w = (.375)(\text{[illegible]}) = 5.18\ \#/\text{FT}^2$
$W = (1.309)(5.18) = 6.78\#$

RECTANGULAR DUCT

$A = [8\pi + 2(10+12)]\,6/144 = 2.4004\ \text{FT}^2$
$w = 5.18\#/\text{FT}^2$
$W = (2.4004)(5.18) = 12.43\#$

TRANSITION DUCT

$A = \left(\frac{62.832 + 69.1328}{2}\right) 10/144 = 4.5821\ \text{FT}^2$
$w = 5.18\#/\text{FT}^2$
$W = (4.5821)(5.18) = 23.74\#$

REVERSE DUCT

$A = \left[\frac{8\pi}{2} + 15 + 12 + 15\right] 14/144 = 5.3051\ \text{FT}^2$
$w = 5.18\#/\text{FT}^2$
$W = (5.3051)(5.18) = 27.48\#$

REVERSE DUCT FLANGE

$A_1 = 20(4+5) - \frac{(.785)(\text{[illegible]})}{2} - (12)(4) = 106.88$
$A_2 = 20(12) = 240$
$A_3 = 2(16)(2) = 64$
$A = 410.88/144 = 2.8533\ \text{FT}^2$
$w = 5.18\#/\text{FT}^2$
$W = (2.8533)(5.18) = 14.78\#$

REVERSE DUCT LOUVRES (HORIZ.)

$A = 3(2)(13)/144 = .5417\ \text{FT}^2$
$w = 2.592\#/\text{FT}^2$
$W = (.5417)(2.592) = 1.40\#$

REL. [illegible] (VERT.)

$A = 3(20)(4)/144 = 1.667\ FT^2$
$t = .125''$, $w = (.125)(144)(.096) = 1.728\ ^{\#}/FT^2$
$W = (1.667)(1.728) = 2.88^{\#}$

DUCT CONNECTOR

$A = (20.75)\pi(4)/144 = 1.8108\ FT^2$
$w = 5.18\ ^{\#}/FT^2$
$W = (1.8108)(5.18) = 9.38^{\#}$

TOTALS

$W =$	6.78	× 1.50/2.66	3.82
	12.43	"	7.01
	23.74	"	13.39
	27.48	"	15.50
	14.78	"	8.33
	1.40	× 1.0	1.40
	2.88	"	2.88
	9.38	"	9.38
	98.87		61.71
	CONV.		PLASTIC

BLADES

x	c	t	a	T.M.	$f(v)$
.20	10.70	.725	5.51	½	2.76
.30	12.25	.638	6.55	1	5.55
.40	13.70	.550	5.35	1	5.35
.50	14.94	.463	4.91	1	4.91
.60	16.00	.375	4.26	1	4.26
.70	16.90	.288	3.46	1	3.46
.80	17.50	.201	2.50	1	2.50
.90	17.85	.113	1.43	1	1.43
1.00	18.00	.080	1.02	½	.51
					30.73

$$\nabla = 3(1)(30.73) = 92.19 \text{ in}^3$$
$$w = .096 \text{ \#/in}^3$$
$$W = (92.19)(.096) = 8.85 \text{ \#}$$

HUB

125.60

$$\nabla = (10)(.785)(4)^2 - (6)(.785)(1.25)^2 - (4)(.785)(3.25)^2 = 78.01 \text{ in}^3$$
$$w = .096 \text{ \#/in}^3$$
$$W = (78.01)(.096) = 7.49 \text{ \#}$$

[illegible]

[illegible] 8.85
7.49

[illegible] [illegible] 8.60 #

[illegible] PLAST.

22962

PLATE

$A = (20)(26)/144 = 3.611\ FT^2$
$t = .25''$, $w = (.25)(144)(.096) = 10.37\ ^{\#}/FT^2$
$W = (3.6111)(10.37) = 37.44^{\#}$

FLANGES

$A = \left[\left(\frac{12+2}{2}\right)9 + 2(12)(2)\right]2/144 = 1.5417\ FT^2$
$t = .25''$, $w = (.25)(144)(.28) = 10.08\ ^{\#}/FT^2$
$W = (1.5417)(10.08) = 15.54^{\#}$

STOCKS

$\ell = (8.0 + 1.5)/12 = .7917'$
$w = (.785)(2)^2(12)(.28) = 10.5504\ ^{\#}/IN$
$W = (.7917)(10.5504) = 8.35^{\#}$

TOTALS

$W =$ 37.44	x 1.-0/2.66	19.71
15.54	x 1.00	15.54
[illegible]	"	8.35
61.3[illegible] $^{\#}$		43.60 $^{\#}$
[illegible]		COMPOSITE

[illegible]

$\ell = 10'$

$w = (.785)(2)^2(12)(.096) = 3.6173 \text{ \#/FT}$

$W = (10)(3.6173) = 36.17\text{\#}$

265.77

Strut (alum) (lower ... support)

$A = (18.5)(6.50)/144 = .8351$

$w = (.50)(144)(.096) = 6.91\ \#/ft^2$

$W = (.8351)(6.91) = 5.77\ \#$

Tiller (alum)

$\nabla = (4)(.785)(4^2 - 2^2) + 2(2)(3)(.5) = 31.68\ in^3$

$w = .096\ \#/in^3$

$W = (31.68)(.096) = 3.04\ \#$

Bearings, Spacers, Webs (Teflon)

$\nabla = (3.6)(.785)(6.50)^2 + (5.50 - 3.60)(.785)(4)^2 - (5.5)(.785)(2)^2 = 125.99\ in^3$

$w = (.096)\left(\frac{1.50}{2.66}\right) = .054\ \#/in^3$

$W = (125.99)(.054) = 6.80\ \#$

Totals

$W = $ 5.77

3.04

6.80

15.61 #

Sta	ord.	a	T.M.	[illegible]
0	11.5	236	½	118
1	9	216	1	216
2	6	144	1	144
3	3.25	98	1	98
4	1.25	30	1	30
5	0	0	½	0
				606

$\nabla = (12)(606)/1728 = 4.21\ FT^3$
$w = 6^{\#}/FT^3$
$W = (4.21)(6) = 25.26^{\#}$

DUCT FAIRING

$\nabla = (12)\left[(24)(24) - (.785)(20)^2\right](60) = 2107\ IN^3 = 1.22\ FT^3$
$w = 30^{\#}/FT^3$
$W = (1.22)(30) = 36.60^{\#}$

[illegible]

[illegible]

$61.86^{\#}$

[illegible]

$W = 223.58 - 10.08 - [illegible] - [illegible] = 249.54^{\#}$

$\nabla = 249.54/166 = 1.50\ FT^3$

$B = (64)(1.50) = 96^{\#}$

[illegible]

$W = 10.08 + 8.35 = 18.43^{\#}$

$\nabla = 18.43/484 = .0381\ FT^3$

$B = (.0381)(64) = 2.44^{\#}$

<u>INLET FAIRING</u>

$\nabla = 4.21\ FT^3$

$B = (4.21)(64) = [illegible]^{\#}$

<u>DUCT FAIRING</u>

$\nabla = 1.22\ FT^3$

$B = (1.22)([illegible]) = [illegible]$

<u>TOTALS</u>

$B =$ 96.00
[illegible]
[illegible]
[illegible]8.08

445.96 #

D[illegible] Shaft & Joint[illegible]	42.[illegible]
Dr[illegible] Shaft Cover	17.00
A[illegible] U-Joint Support	2.90
A[illegible] Drive Shaft	34.04
W[illegible] U - Assembly	304.[illegible]
F[illegible] Duct	18.50
M[illegible]	11.00
	435.3[illegible] [illegible]

WEIGHT SUMMARY

	CONVENTIONAL CONST.		COMPOSITE CONST.		
	WT.	MAT'L	WT.	MAT'L	
PROP. DUCT	[illegible]	ALUM. (5086-H116)	31.16	POLYESTER - GLASS CLOTH + ALUM. [illegible]	
RUDDER DUCT	[illegible]	" "	61.71	" " "	
PROPELLER	16.34	ALUM (356-T?)	8.60	POLYCARB - 30% GLASS	
RUDDER	[illegible]	ALUM (6061-T6)	43.60	POLYESTER - GLASS CLOTH [illegible]	
SHAFT	[illegible]	" "	36.17	ALUM. (6061-T6)	
MISC.	[illegible]		15.61	[illegible], ALUM. [illegible]	
			196.85		[illegible] = [illegible]
FAIRINGS	61.86	FOAM	61.86	FOAM	
	345.44		258.71		
BUOYANCY	[illegible]		445.96		167.97 [illegible]
	- 100.52		- 187.25		+ 267.40

EXISTING PUMP

- STATIC PERFORMANCE
- CRUISE PERFORMANCE
- COMPARISON

EXISTING [illegible]

STATIC PERFORMANCE

$T_{STAT} = 3025^{\#}$ [illegible] (KNOWN)

$A_J = (.785)(10.88)^2/144 = .6453\ FT^2$ (KNOWN)

$V_J = \sqrt{\frac{T_{STAT}}{\rho A_J}} \quad \sqrt{\frac{3025}{2(.6453)}} = 48.41\ FT/SEC$

$Q = V_J A_J = (48.41)(.6453) = 31.24\ FT^3/SEC = 14020\ GPM$

$H_P = HL_J + HL_I - H_O$

$HL_J = \frac{V_J^2}{2g} = \frac{(48.41)^2}{2(32.2)} = 36.39'$

$HL_I = .000533\,Q^2 = (.000533)(31.24)^2 = .52'$

$H_O = 0$ (STATIC)

$H_P = 36.39 + .52 - 0 = 36.91'$

$WHP = \frac{\rho g\,Q\,H_P}{550} = \frac{(2)(32.2)(3[illegible])}{5[illegible]} = 135$

$SHP = 200$

$\eta_{PUMP + NOZZLE} = \frac{WHP}{SHP} = \frac{1[illegible]}{[illegible]} = .[illegible]$ SEEMS LOW PERHAPS PUMP IS NOT USING ALL AVAILABLE POWER

CRUISE PERFORMANCE

$V_O = 8\ MPH = 11.76\ FT/SEC$

$RPR = .50$

$H_O = (RPR)\frac{V_O^2}{2g} = (.50)\frac{(11.76)^2}{2(32.2)} = 1.07'$

$H_J = H_P - HL_I + H_O = 36.91 - .52 + 1.07 = 37.46'$

$V_J = \sqrt{2g\,H_J} = \sqrt{2(32.2)(37.46)} = 49.12\ FT/SEC$

$Q = V_J A_J = (49.12)(.6453) = 31.70\ FT^3/SEC$

$T = \rho Q\,(V_J - V_O) = 2(31.70)(49.12 - 11.76) = 2369^{\#}$

$P.C. = \frac{T V_O}{550\ SHP} = \frac{(2369)(11.76)}{(550)(200)} = .2533$

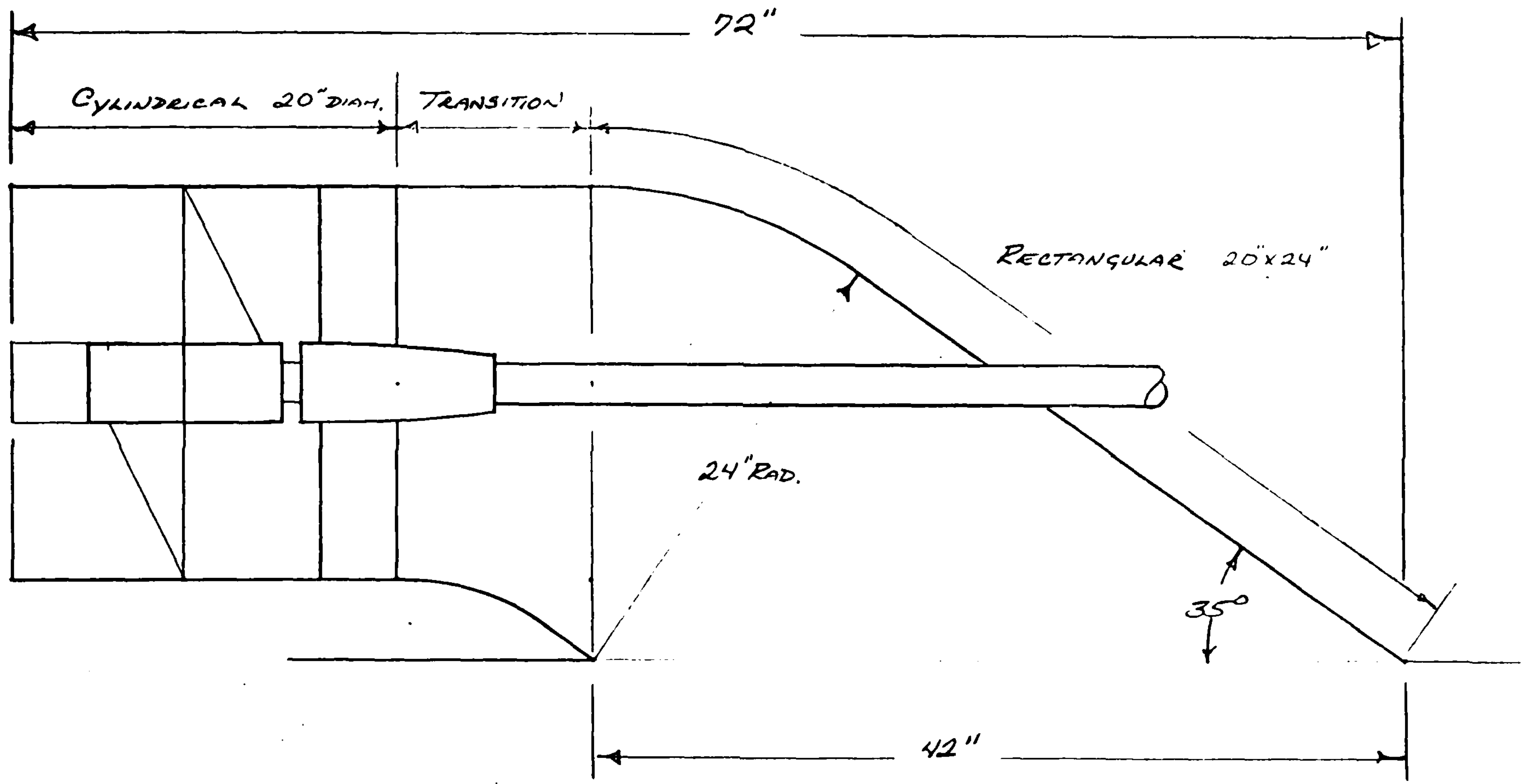
COMP[illegible]

	F[illegible] SYST[illegible]	F[illegible]
THRUST AT [illegible]	284[illegible]#	2359#
P.C. AT 8MPH	.30	.25
FLOW RATE AT 8 [illegible]	35[illegible]46 GPM	14020 GPM
STATIC THRUST	4278#	3025#
DRY WEIGHT	2[illegible]# CONV. CONST.	435#
	197# COMP. CONST	

APPENDIX B

OBJECTIVES:

- Determine performance, weight and dimensional characteristics of a propulsion pump, about the same size as the PJ-16, for use in a high-speed (20 mph) amphibian.
- Use simple "propeller-in-tube" approach.
- Limit blade area ratio to that available in existing propeller series.
- Investigate use of composite materials.



SKETCH OF 20 INCH DIAMETER PUMP

PERFORMANCE CHARACTERISTICS

- POWER LIMITS
- CAVITATION LIMITS
- SYSTEM PERFORMANCE
- SYSTEM OPERATION
- DERIVATIONS + REF. MATL.

CALCULATION NOTES

D = PROP. DIAM. = 20" = 1.667'

A_P = PROP. DISC. AREA = $.785\left[(1.667)^2-(.333)^2\right] = 2.0942\ FT^2$

A_I = INLET AREA = $(1.667)(2.00) = 3.333\ FT^2$

J_{CYL} = ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE

J_e = ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE

$$\frac{J_{CYL}}{J_e} = \frac{A_I}{A_P} = \frac{3.333}{2.0942} = 1.5915$$

P/D = PROP. PITCH DIAM. RATIO

$J_{CYL} = f\left(P/D, \frac{J_{CYL}}{J_e}\right) = .905$ (VON LAMMEREN - FIG. 1)

$J_e = J_{CYL} \Big/ \left(\frac{J_{CYL}}{J_e}\right) = .905/1.5915 = .5686$

η_0 = PROPULSION EFFICIENCY OF PROP-TUBE COMB. = $f(J_e, P/D)$ (FIG. 2)

η_{RR} = PROP. RELATIVE ROTATIVE EFFICIENCY = .95 (GUESS - TUNNEL PAR...

η_P = PUMP EFFICIENCY = $\frac{J_{CYL}\ \eta_0\ \eta_{RR}}{J_e}$

SHP = PUMP INPUT POWER

Q = FLOW RATE [illegible]

PUMP [illegible] = $\frac{550\ SHP\ \eta_P}{\rho g\ Q}$

Power Limit

Calculation

D	A_P	A_I	$\frac{J_{cyl}}{J_e}$	[illegible]	J_{cyl}	J_e	η_o	η_{rr}	η_p	SHP	Q	H_{avail}
1.667	2.0942	3.333	1.5915	1.00	.995	[illegible]	.51	.95	.77	500	70	46.97
											80	41.10
											90	36.53
											100	32.88
											110	29.89
										400	70	37.58
											80	32.88
											90	29.23
											100	26.30
											110	23.91
										300	70	28.18
											80	24.66
											90	21.92
											100	19.73
											110	17.93
										200	70	18.79
											80	16.44
											90	14.61
											100	13.15
											110	11.96

[illegible]

Calculation Notes

D = PROP. DIAM. = 20" = 1.667'

A_P = PROP. DISC AREA = $.785\left[(1.667)^2 - (.333)^2\right] = 2.0942\ FT^2$

A_I = INLET AREA = $(1.667)(2.00) = 3.333\ FT^2$

J_{CYL} = ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE

J_E = ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE

$$\frac{J_{CYL}}{J_E} = \frac{A_I}{A_P} = \frac{3.333}{2.0942} = 1.5915$$

P/D = PROP. PITCH DIAM. RATIO

$J_{CYL} = f\left(P/D, \frac{J_{CYL}}{J_E}\right)$ (VON LAMERRN – FIG. 1)

Q = FLOW RATE ~ FT³/SEC

n = PROP. SPEED = $\dfrac{Q}{A_P\, J_{CYL}\, D}$

H_{LI} = INLET HEAD LOSS = $.000234\ Q^2$ (DERIVATION #1)

V_I = INLET VELOCITY = Q/A_I

$V_{.7TAN}$ = TANGENTIAL VELOCITY AT .7 RAD. OF PROP. = $.7\pi D n$

$V_{.7R}$ = TOTAL VELOCITY AT .7 RAD. OF PROP. = $\sqrt{V_I^2 + V_{.7TAN}^2}$

H_{ATM} [illegible] DUE TO [illegible]

H_E = HEAD DUE TO ELEVATION

H_V = HEAD DUE TO VAP. PRESS.

CALCULATION NOTES: (CONT.)

V_0 = FREE STREAM VELOCITY = CRAFT SPEED

RPR = RAM PRESS. RECOVERY RATIO = .70 (JACUZZI – FIG. 3)

H_0 = RAM HEAD RECOVERED = $(RPR)\frac{V_0^2}{2g}$

H_{I_S} = INLET STATIC HEAD (ABOVE VAP. PRESS.) = $H_{ATM} + H_2 - H_V + H_0 - HL_I - \frac{V_I^2}{2g}$

P_{I_S} = INLET STATIC PRESS. (ABOVE VAP. PRESS.) = $\rho g H_{I_S}$

$\sigma_{.7R}$ = LOCAL CAV. NO. AT .7RAD. = $\frac{P_{I_S}}{\rho/2 \, V_{.7R}^2}$

T_{CMAX} = PROP. LOAD COEF. AT CAV. LIMIT = $.7\,\sigma_{.7R}$ (GAWN)

EAR = PROP. EXPANDED AREA RATIO = 1.18 (MAX. AVAILABLE)

PAR = PROP. PROJECTED AREA RATIO = 1.01 (DERIVATION #3)

$H_{P_{CAV.}}$ = PUMP HEAD RISE AT CAV. LIMIT = $\frac{(T_{CMAX})(PAR)(V_{.7R})^2(\rho/2)}{\rho g}$

CAVITATION LIMIT

CALCULATION

D	A_p	A_s	J_{cyc}/J_a	P/D	J_{cyc}	Q	n	H_{LS}	V_I	V_{Rem}	$V_{.7R}$	$V_I^2/2g$	H_{atm}	H_s	H_V	V_o	RPR	H_o	H_{S_o}	P_{SL}	$\sigma_{.7R}$	τ_{cmr}	EAR	PAR	$H_{cav.}$
1.667	2.0746	3.333	1.5915	1.00	.905	70	22.16	1.15	21.00	81.24	83.9	6.85	33.06	1	1	14.70	.70	[illegible]	27.43	1766	.2508	.1756	1.18	1.01	19.39
						80	25.32	1.50	24.00	92.82	95.67	8.94							24.99	1609	.1751	.1226			17.67
						90	28.49	1.90	27.00	104.44	107.87	11.32							22.21	1430	.1229	.0860			15.69
						100	31.65	2.34	30.00	116.03	119.65	13.98							19.11	1231	.0857	.0600			13.52
						110	34.82	2.83	33.00	127.65	131.65	16.91							15.69	1010	.0581	.0407			11.10
						70										22.05	.70	[illegible]	30.36	1755	.2777	.1944			21.47
						80													27.92	1798	.1956	.1369			19.33
						90													25.14	1619	.1391	.0974			17.77
						100													22.04	1419	.0988	.0692			15.59
						110													18.62	1199	.0690	.0483			13.17
						70										29.40	.70	[illegible]	34.48	2221	.3154	.2208			24.38
						80													32.04	2063	.2245	.1572			22.66
						90													29.26	1884	.1619	.1133			20.68
						100													26.16	1685	.1173	.0821			18.50
						110													22.74	1464	.0842	.0589			16.06
						70										36.75	.70	11.65	39.76	2561	.3637	.2546			28.11
						80													37.32	2403	.2614	.1830			26.38
						90													34.54	2224	.1911	.1338			24.42
						100													31.44	2025	.1410	.0987			22.23
						110													28.02	1804	.1038	.0727			19.82
						50	15.83	.58	15.00	58.03	59.94	3.49	33.06	1	1	0		0	29.01	1868	.5199	.3639			20.50
						60	18.99	.84	18.00	69.62	71.91	5.03							27.21	1752	.3383	.2372			19.24
						70	22.16	1.15	21.00	81.24	83.91	6.85							25.08	1615	[illegible]	[illegible]			17.73

STATIC OPERATION

CALCULATION NOTES

- REQUIRED PUMP HEAD

V_o = CRAFT SPEED

Q = FLOW RATE ~ FT^3/SEC

$H_{PREQ} = .00381 Q^2 - .0109 V_o^2$ (DERIVATION #2)

- ESTIMATED THRUST (POWER LIMIT)

V_o = CRAFT SPEED

SHP = PUMP INPUT POWER

Q_{EQ} = EQUILIBRIUM FLOW RATE @ $H_{PAVAIL} = H_{PREQ}$.

A_J = JET AREA = A_P = 2.0942 FT^2

V_J = JET VELOCITY = Q_{EQ}/A_J

T = THRUST = $\rho Q (V_J - V_o)$

P.C. = PROPULSIVE [illegible] = $\frac{T V_o}{[illegible]}$

- ESTIMATED THRUST ([illegible])

V_o = CRAFT [illegible]

Q_{EQ} = EQUILIBRIUM [illegible] @ $H_{PCAV} = H_{PREQ}$

A_J = JET AREA = A_P = 2.0942 FT^2

V_J = JET VELOCITY = Q_{EQ}/A_J

T = THRUST = $\rho Q (V_J - V_o)$

H_{PEQ} = EQUILIBRIUM HEAD RISE @ $H_{PCAV} = H_{PREQ}$

η_p = PUMP EFFICIENCY = .77 (POWER LIMIT CALC.)

SHP = PUMP INPUT POWER = $\frac{\rho g H_{PEQ} Q_{EQ}}{550 \eta_p}$

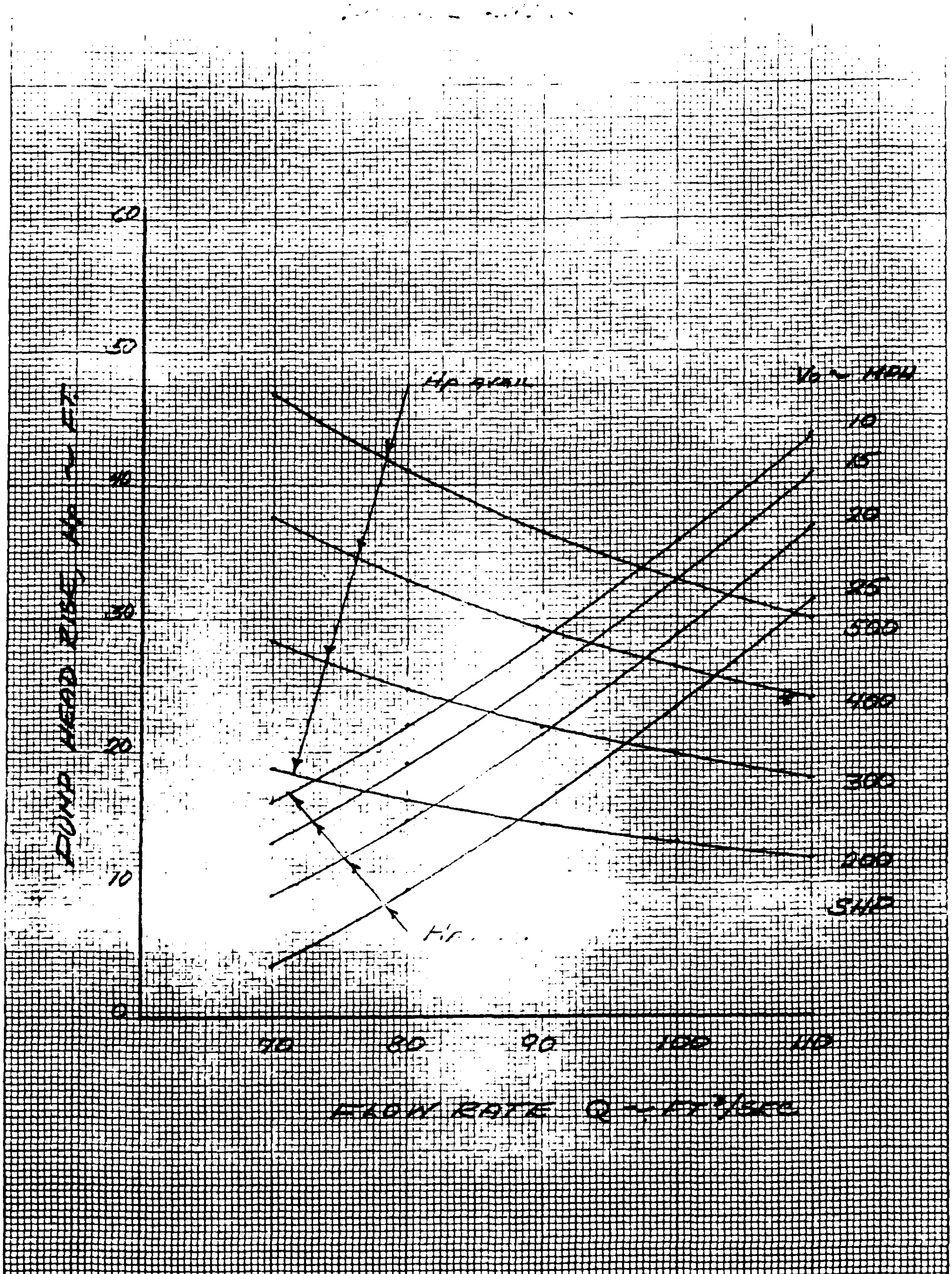
P.C. = PROPULSIVE COEF. = $\frac{T V_o}{550 SHP}$

SYSTEM PERFORMANCE

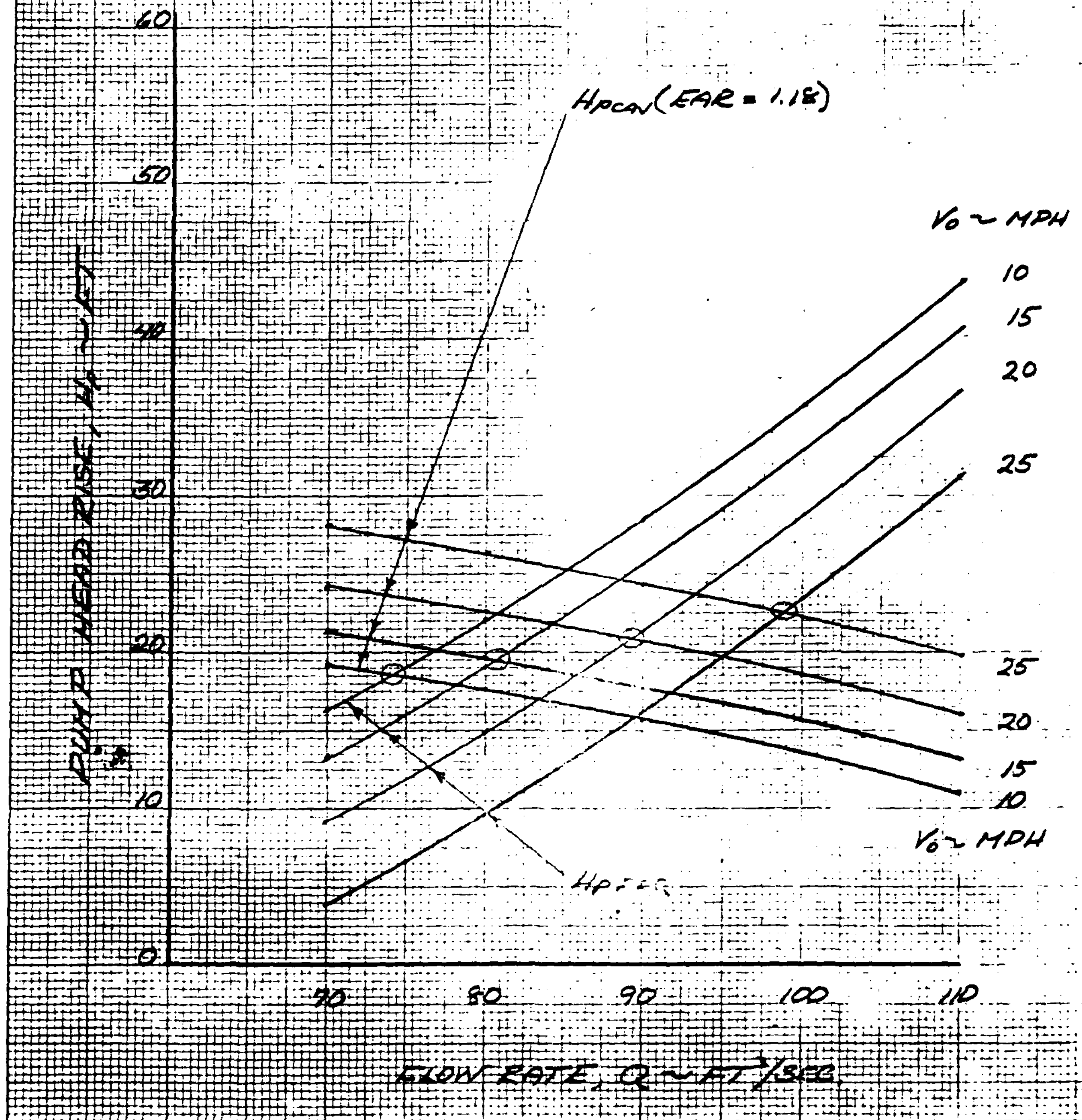
CALCULATIONS

- REQUIRED PUMP HEAD

V_o	Q	$H_{P\,REQ}$
14.70	70	16.31
	80	22.03
	90	28.51
	100	35.74
	110	43.75
22.05	70	13.37
	80	19.08
	90	25.56
	100	32.80
	110	40.80
29.40	70	9.25
	80	14.96
	90	21.44
	100	28.68
	110	36.68
36.75	70	3.95
	80	9.66
	90	16.14
	100	23.38
	110	31.38



60
50
40
30
20
10
0
PUMP HEAD RISE, Hp ~ FT.
Hp avail
Vo ~ MPH
10
15
20
25
500
400
300
200
SHP
70
80
90
100
110
FLOW RATE Q ~ FT³/SEC



60
50
40
30
20
10
0
70
80
90
100
110
PUMP HEAD RISE, H_p ~ FT
FLOW RATE, Q ~ FT³/SEC
H_{pCAV} (EAR = 1.18)
V_0 ~ MPH
10
15
20
25
25
20
15
10
V_0 ~ MPH

SYSTEM P[illegible]

CALCULATIONS (CONT.)

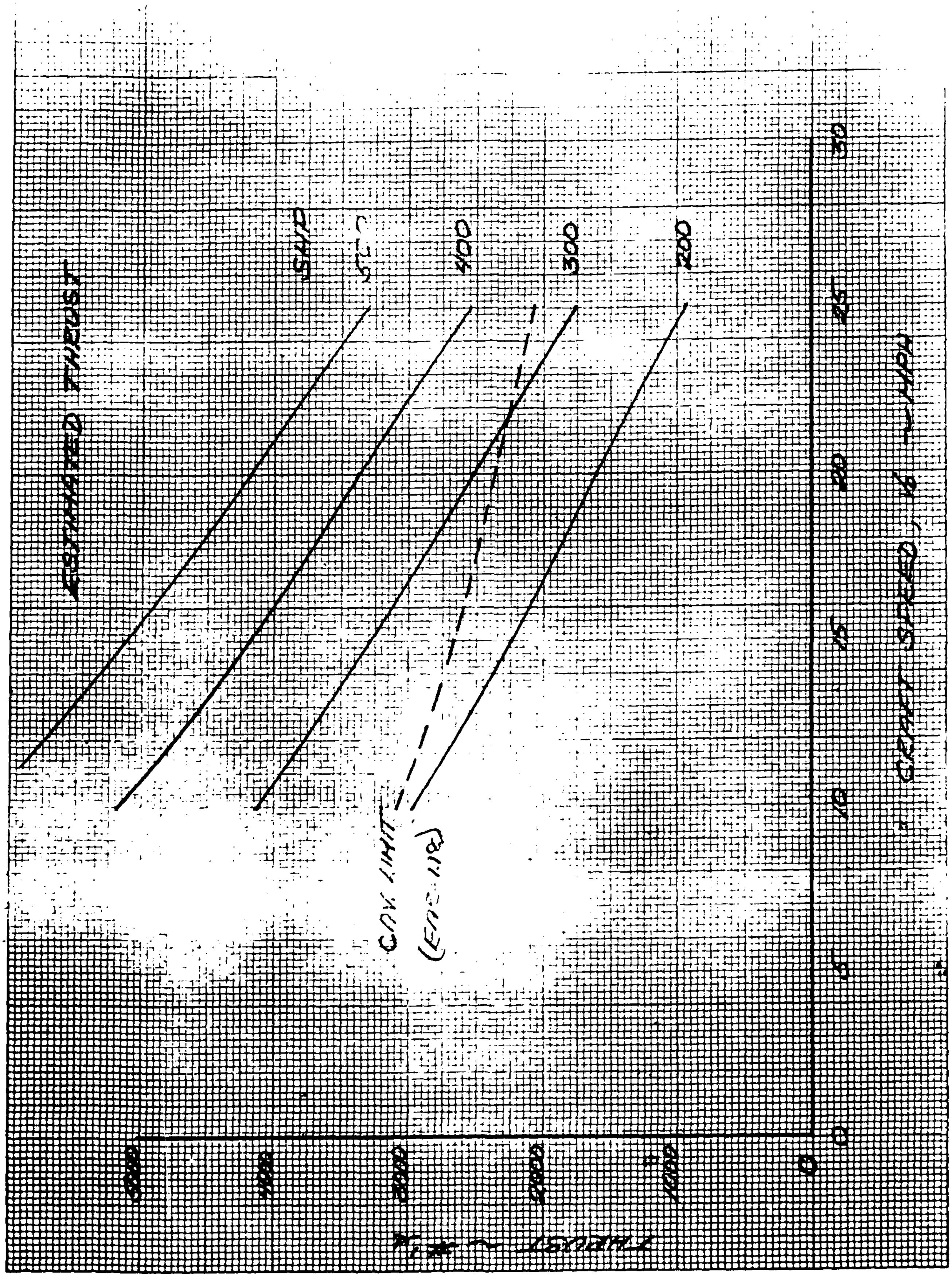
• ESTIMATED THRUST (POWER LIMIT)

V_o	SHP	Q	A_J	V_J	T	P.C.
14.70	500	97.4	2.0942	46.51	6197	.3313
	400	90.7		43.31	5190	.3468
	300	83.0		39.63	4138	.3687
	200	73.2		34.95	2965	.3962
22.05	500	100.0		47.75	5140	.4121
	400	93.2		44.50	4185	.4195
	300	86.2		41.16	3295	.4403
	200	76.8		36.67	2246	.4502
29.40	500	103.8		49.57	4187	.4476
	400	97.7		46.65	3371	.4505
	300	90.7		43.31	2523	.4496
	200	82.0		39.16	1601	.4279
36.75	500	108.8		51.95	3308	.4421
	400	102.9		49.14	2550	.4260
	300	96.2		45.94	1768	.3938
	200	88.2		42.12	947	.3164

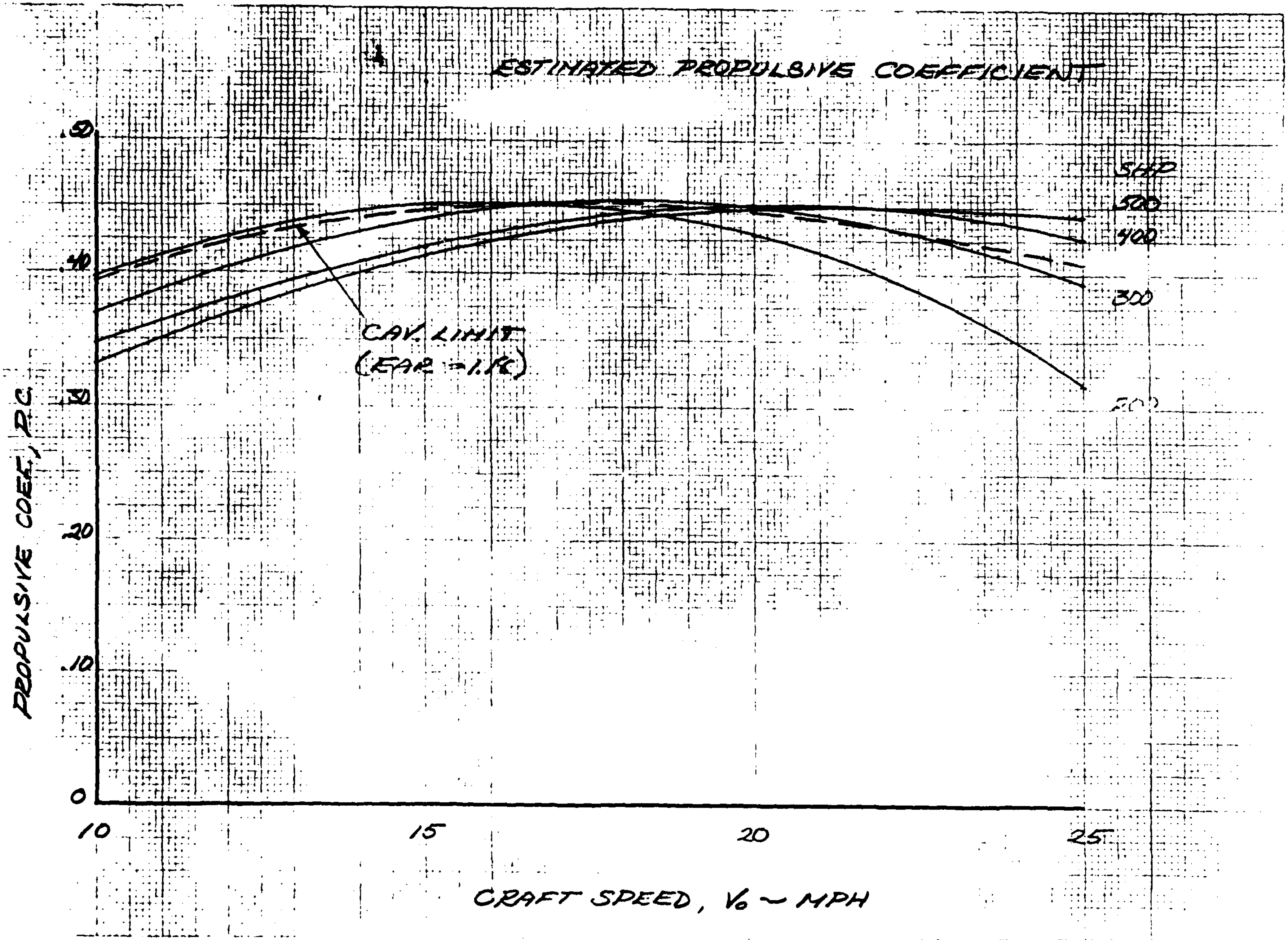
CALCULATIONS (CONT.)

- ESTIMATED THRUST ([illegible])

V_0	Q_{EQ}	A_J	V_J	T	H_{PEQ}	η_P	SHP	PC
14.70	74.3	2.0942	35.48	3088	18.6	.77	210	.3930
22.05	80.7		38.54	2661	19.5		239	.4464
29.40	89.3		42.64	2365	20.9		284	.4451
36.75	98.8		47.18	2061	22.5		338	.4074



ESTIMATED THRUST
SHP
500
400
300
200
CAV. LIMIT
CRAFT SPEED, Vk ~ MPH
0
5
10
15
20
25
30
THRUST ~ T, #
0
1000
2000
3000
4000
5000


ESTIMATED PROPULSIVE COEFFICIENT
SHP
500
400
300
200
CAV. LIMIT
(EAR = 1.16)
PROPULSIVE COEF., PC
.50
.40
.30
.20
.10
0
10
15
20
25
CRAFT SPEED, V_0 ~ MPH

(Static Operation)

Required Pump Head

$H_{PREQ} = .00281 Q^2$

Q	H_{PREQ}
50	9.52
60	13.72
70	18.67

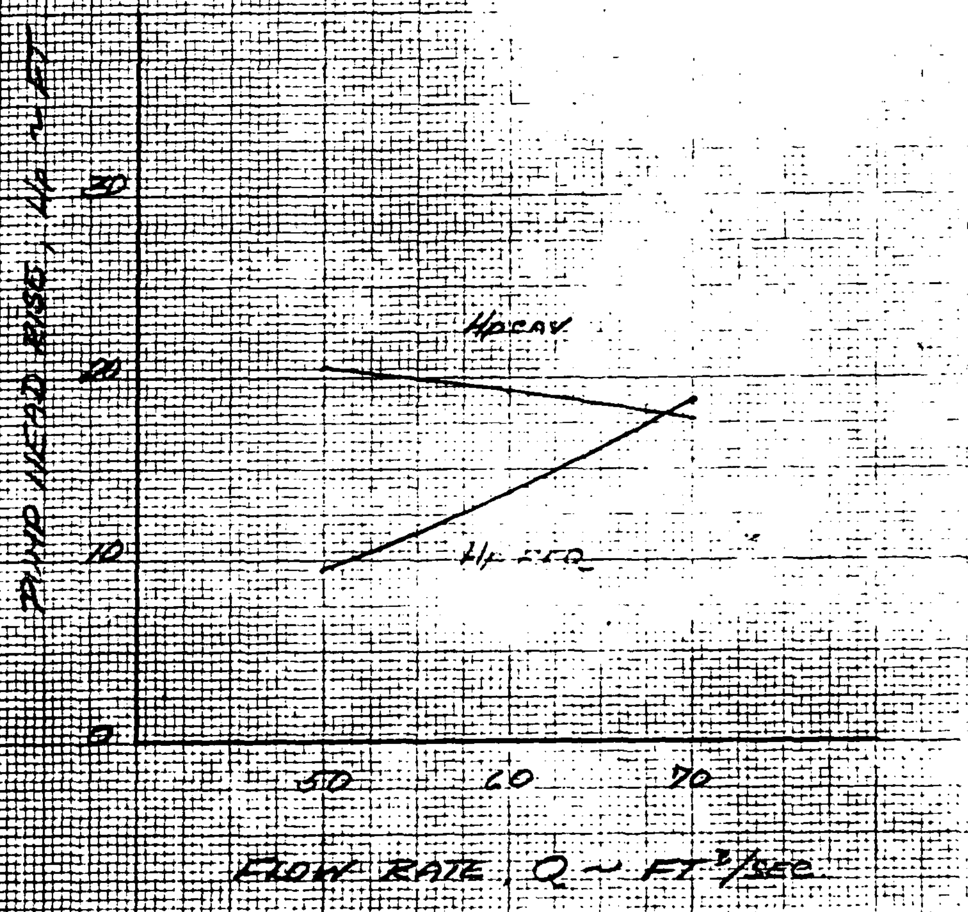
Cavitation Limit (see cavitation limit calcs.)

Q	H_{PCAV}
50	20.60
60	19.24
70	17.73

Estimated Thrust

Q_{EQ}	A_J	V_J	$T_{STAT.}$	H_{ATMOS}	H_Z	H_{LI}	A_I	V_I	$\frac{V_I^2}{2g}$	H_{IS}	P_{IS}
68.7	2.0942	32.80	4507	33.06	1	1.10	3.333	20.61	6.60	26.38	1699

USE 70 [illegible]
[illegible] PRESSURE ([illegible])


PUMP HEAD RISE, Hp ~ FT
30
20
10
0
Hpcav
50
60
70
FLOW RATE, Q ~ FT³/SEC

DERIVATIONS & [illegible]

REF. MATL.

VON LAMEREN FIG. 1 } MODEL [illegible] FLOW AXIAL CYL.
" FIG. 2

JACUZZI FIG. 3 MODEL [illegible] INLET

DERIVATIONS

#1 ESTIMATED INLET & CASING LOSSES

#2 REQUIRED PUMP HEAD RISE

#3 PROPELLER BLADE AREA

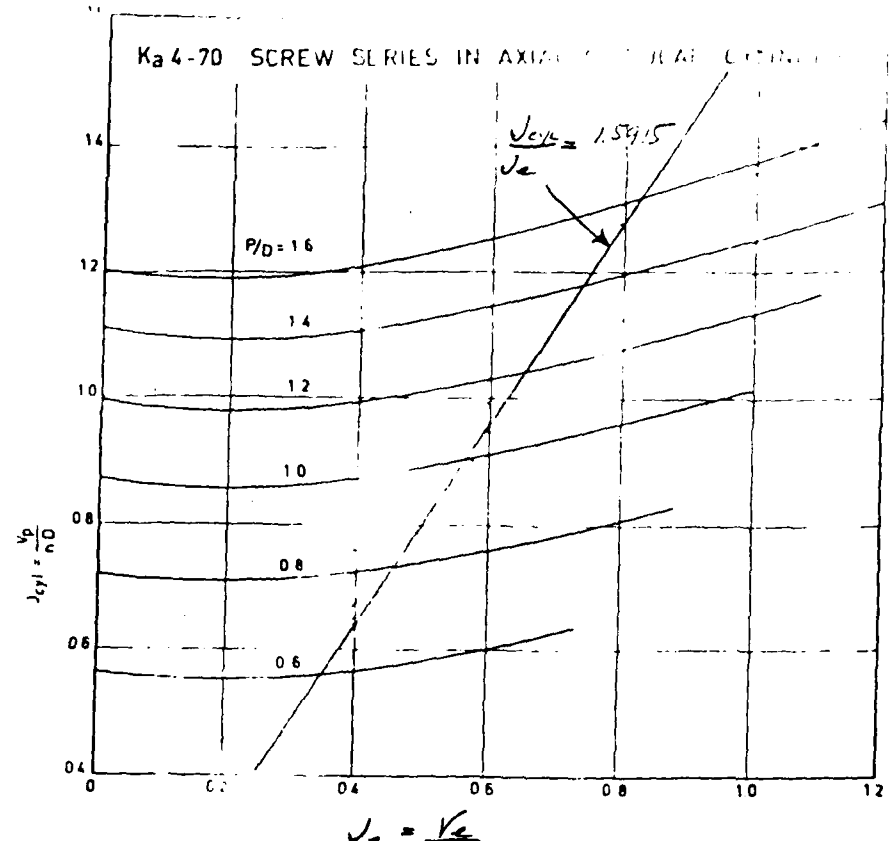


Fig. 29 Relation between velocity of "screw + cylinder" combination and velocity in cylinder

Fig. 1

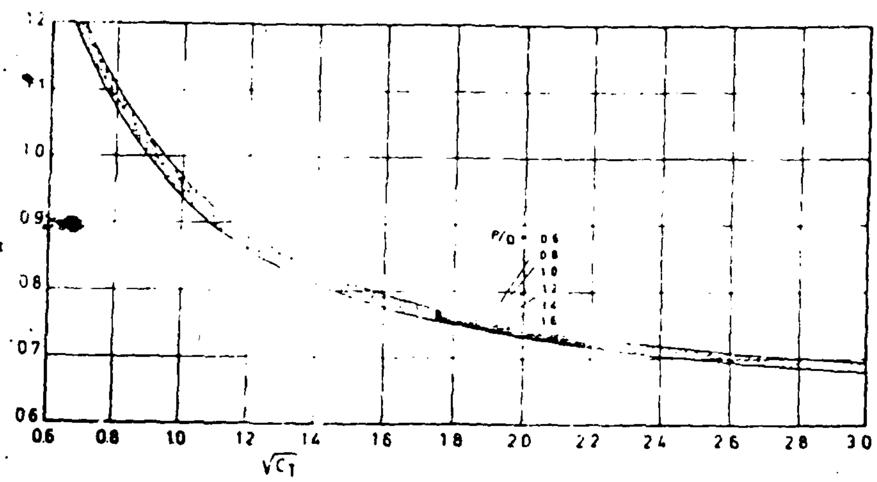


Fig. 30 Relation between thrust coefficient C_T and thrust ratio τ of nozzle no. 19a

figure have been obtained by substituting nozzles with different length-diameter ratios by systems of annular vortexes and calculating the induced velocities in the screw disk.

If the radial displacement of the streamlines is small, we can consider the streamlines as lying approximately on cylindrical planes. If internal friction and turbulence are neglected, the radial

Von Lammeren

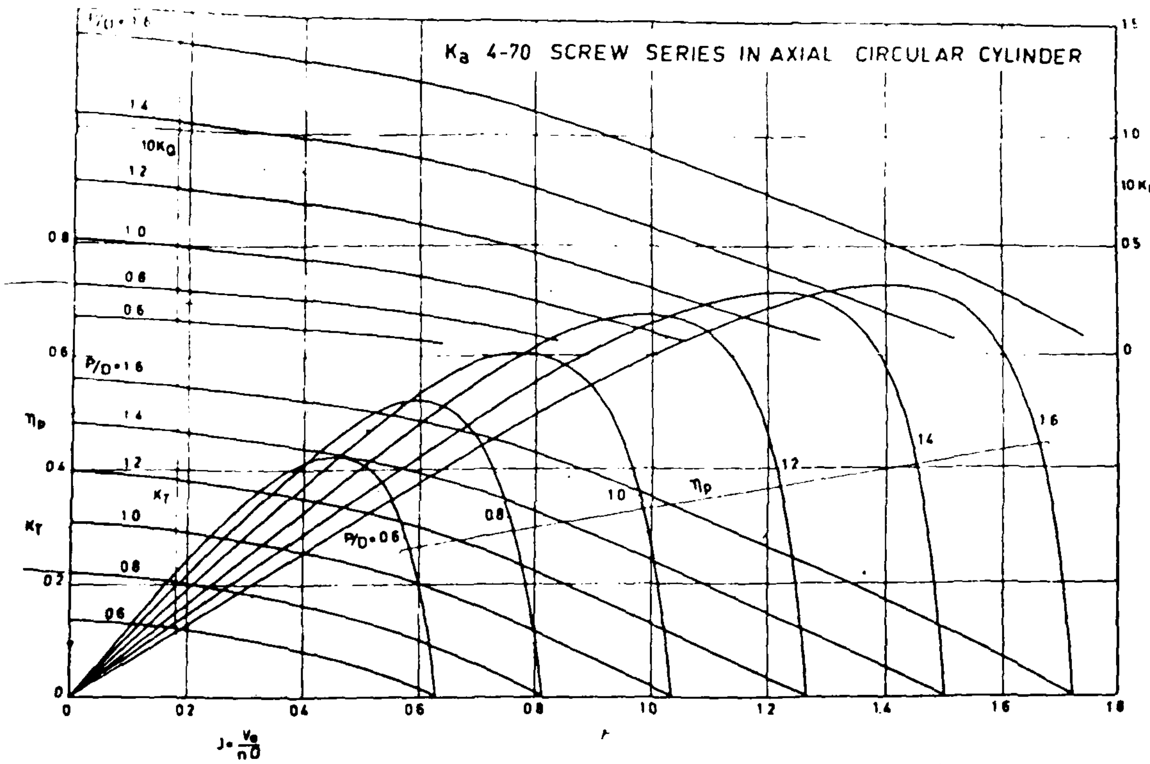


Fig. 28 Results of open-water tests with Ka 4-70 screw series in an axial cylinder

F_IG 2

been obtained from the experiments with the Ka 4-70 screw series in an axial circular cylinder and from the application of the momentum theorem.

From the comparison of the axial velocities obtained with these methods, we see that

1 The velocities agree reasonably well at high loadings of the ducted propeller system ($C_T > 1$).

2 The difference between the axial velocities becomes very large at low loadings ($C_T < 1$).

In regard to the second conclusion, the following remark may be made. From Fig. 13 it can be seen that the nozzle drag due to friction becomes substantial at low loadings of the ducted-propeller system. Then, it is no longer permitted to neglect the effect of friction on the force action between nozzle and fluid.

The design of a screw in a nozzle may now be carried out as follows:

With given thrust T or power P, intake velocity V_a, and number of revolutions n, the B_p and consequently the optimum diameter coefficient D can be determined with the aid of open-water test results of the nozzle considered, in combination with a systematic screw series (see, for instance, Fig. 24). In addition, the thrust coefficient C_T and the propeller thrust-total thrust ratio τ can be determined. With the aid of the experiments of the systematic screw series in the axial circular cylinder or using the momentum theorem, the axial velocity V_P in the way of the screw can be found. In addition, the mean axial velocity in the vicinity of the screw due to the nozzle action, U_N, and due to the screw action U_P, can be calculated.

The pressure difference created by the screw becomes

$$\Delta p = \frac{T_p}{\frac{\pi}{4}(D^2 - d_h^2)}$$

In order to avoid an excessive loading of the inner radii of the screw blades, the usual assumption for axial pumps that the head is constant for all radii is abandoned. The following radial $\Delta p(r/R)$ distribution is suggested for the screws in nozzle no. 19*a*:

$$\Delta p(r/R) = [4.88 - 4r/R] \cdot [r/R - 0.133] \Delta p$$

The radial distribution of the axial and tangential velocities at the screw may be approximated as follows:

A reasonable radial distribution of the axial velocities due to the nozzle action can be determined from Fig. 32. The results given in this

Von Lamern

B-21

COMPARISON OF RECTANGULAR AND ELLIPTICAL
INLET RAM RECOVERY VARIATIONS WITH
INLET VELOCITY RATIO

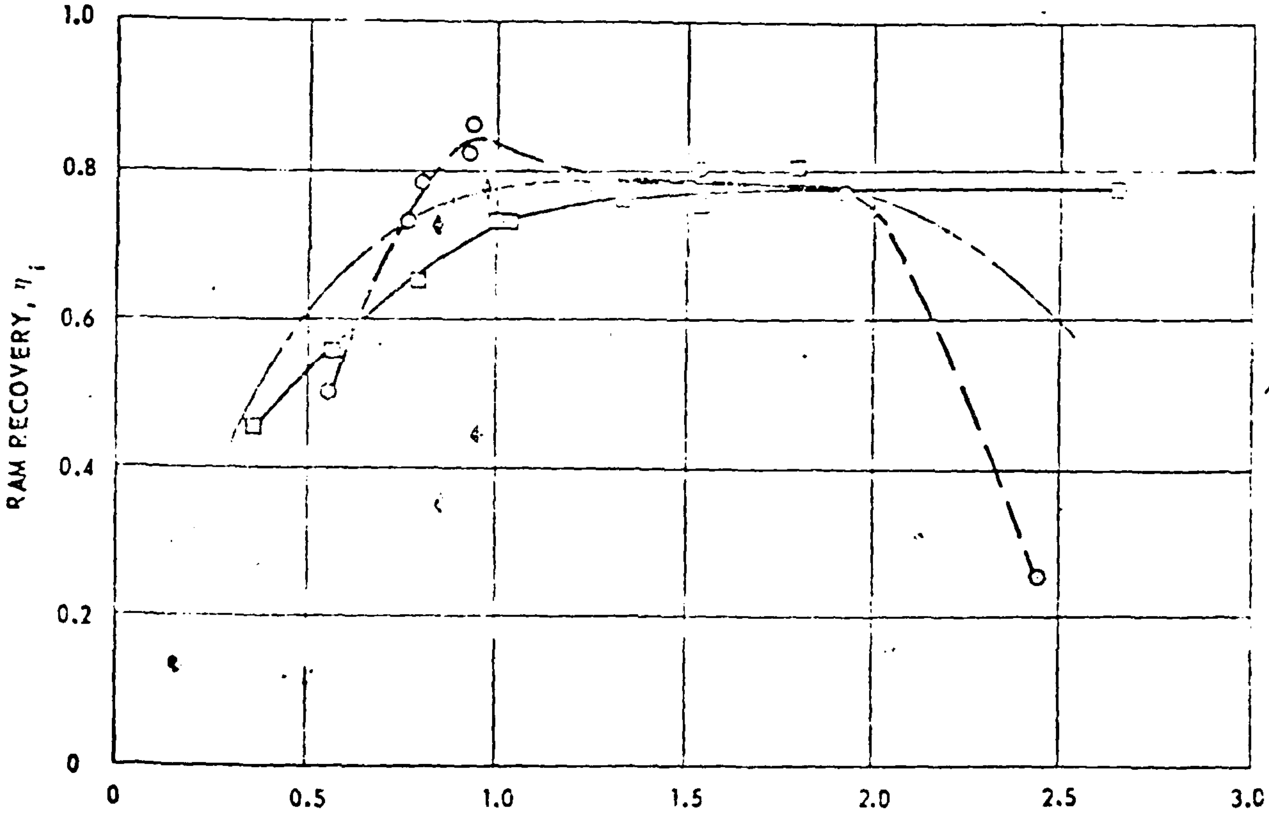
(Laboratory Water Channel Test of 2-inch Eye Diameter Waterjet Inlet Models)

SYM	CONFIGURATION
□	RECTANGULAR
○	ELLIPTICAL – 0.3 IN. AFT LIP RADIUS

– – – – Estimated Performance of Jacuzzi Inlet Configuration

$$\eta_i = 1 - \frac{(P_{T_\infty} - \overline{P}_i)}{q_\infty}$$

♦ Measured Performance of 28HJ Inlet



AVG. INLET EXIT PLANE VELOCITY / FREE STREAM VELOCITY $\frac{\overline{V}_i}{V_\infty}$

FIG. 3

B-22 Jacuzzi

INTAKE FRICTION & BEND

Q = NOMINAL FLOW RATE = 100 FT^3/SEC

A_I = INLET AREA = $(1.667)(2.00) = 3.333$ FT^2

V_I = INLET VELOCITY = $Q/A_I = 100/3.333 = 30$ FT/SEC

d_e = EQUIV. INLET DIAM. = $\dfrac{4A_I}{2(1.667+2.00)} = \dfrac{4(3.333)}{2(1.667+2.00)} = 1.8182'$

$R_e = \dfrac{V_I d_e}{1.24\times10^{-5}} = \dfrac{(30)(1.8182)}{1.24\times10^{-5}} = 4.40\times10^{6}$

e/d_e = RELATIVE ROUGHNESS = $.000005/1.8182 = .00000275$

f = FRICTION FACTOR = .00925

L_I = INTAKE LENGTH = $\sqrt{21^2-12^2} = 17.23'' = 1.44'$

L_B = EQUIV. LENGTH OF BEND = $\left(\frac{L}{D}\right)_B (d_e)\left(\frac{\alpha}{90}\right) = (36)(1.8182)\left(\frac{35}{90}\right) = 25.45'$

L = TOTAL EQUIV. LENGTH = $L_I + L_B = 1.44 + 25.45 = 26.89'$

$HL = f\left(\frac{L}{d_e}\right)\left(\frac{V_I^2}{2g}\right) = (.00925)\left(\frac{26.89}{1.8182}\right)\frac{(30)^2}{2(32.2)} = 1.9118'$

SHAFT

C_D = [illegible] = $1.1\sin^3\left(\frac{\alpha}{2}\right) = 1.1\sin^3\left(\frac{35}{2}\right) = .0299$

V_I = 30 FT/SEC

l = SHAFT LENGTH = 2.667'

d = SHAFT DIAM. = 2" = .1667'

D = SHAFT DRAG = $C_D \frac{\rho}{2} V_I^2 l d = (.0299)\left(\frac{2}{2}\right)(30)^2(2.667)(.1667) = 11.96$ #

$HL = \dfrac{D}{\rho g A_I} = \dfrac{11.96}{(2)(32.2)(3.333)} = .0557'$

[illegible]

TRANSITION

A_T = CROSS SECTION AREA = $\frac{3.333 + 2.0942}{2}$ = 2.7138 FT^2

V_T = VELOCITY = Q/A_T = 100/2.7138 = 36.85 FT/SEC

d_e = EQUIV. DIAM. = $\sqrt{\frac{2.7138}{.785}}$ = 1.8593'

$R_e = \frac{V_T d_e}{1.24 \times 10^{-5}} = \frac{(36.85)(1.8593)}{1.24 \times 10^{-5}} = 5.53 \times 10^6$

ϵ/d_e = RELATIVE ROUGHNESS = .000005/1.8593 = .00000269

f = FRICTION FACTOR = .009

L_T = TRANSITION LENGTH = 10" = .833'

$HL = f\left(\frac{L}{d_e}\right)\left(\frac{V_T^2}{2g}\right) = (.009)\left(\frac{.833}{1.8593}\right)\frac{(36.85)^2}{2(32.2)} = .0850'$

BEARING TUBE

A_P = CROSS SECTION AREA = 2.0942 FT^2

V_P = VELOCITY = Q/A_P = 100/2.0942 = 47.75 FT/SEC

l = TUBE LENGTH = 1.667'

$R_e = \frac{V_P l}{1.24 \times 10^{-5}} = \frac{(47.75)(1.667)}{1.24 \times 10^{-5}} = 6.42 \times 10^6$

C_f = .00315

S = TUBE WETTED SURF = $(1.667)\,\pi\,(.333) = 1.74\ FT^2$

D = TUBE DRAG = $(C_f + .0008)(S)(\rho/2)V_P^2 = (.00315 + .0008)(1.74)(2/2)(47.75)^2 = 15.67$#

$HL = \frac{D}{\rho g A_P} = \frac{15.67}{2(32.2)(2.0942)} = .1162'$

(Continued)

Strut

A_P = CROSS SECTION AREA = 2.0942 FT^2

V_P = VELOCITY = Q/A_P = 100/2.0942 = 47.75 FT/SEC

c = STRUT CHORD = 4" = .333'

$R_e = \frac{V_P c}{1.24 \times 10^{-5}} = \frac{(47.75)(.333)}{1.24 \times 10^{-5}} = 1.28 \times 10^6$

C_f = .00419

t/c = STRUT THICKNESS RATIO = .375/4 = .0938

$C_D = 2(C_f + .0008)(1 + 1.2\,t/c) = 2(.00419 + .0008)(1 + 1.2 \times .0938) = .0111$

S = STRUT PLANFORM AREA = $2(1.667 - .333)(.333)$ = .8884 FT^2

D = STRUT DRAG = $C_D S \frac{\rho}{2} V_P^2 = (.0111)(.8884)(\frac{2}{2})(47.75)^2$ = 22.48#

$HL = \frac{D}{\rho g A} = \frac{22.48}{2(32.2)(2.0942)}$ = .1667'

<u>Total Inlet Loss</u>

Intake Friction + Bend	1.9118
Shaft	.0557
Transition	.0850
Boiling Tube	.1162
Struts	.1667

HL_I = 2.3354'

$k = \frac{HL_I}{Q_{NOM}^2} = \frac{2.3354}{(100)^2}$ = .000234

$HL_I = .000234\ Q^2$

[illegible] ([illegible])

<u>CASING</u>

Q = NOMINAL FLOW RATE = $100\ FT^3/SEC$

$A_p = 2.0942\ FT^2$

$V_p = Q/A_p = 100/2.0942 = 47.75\ FT/SEC$

d = CASING DIAM. = $1.667'$

$Re = \dfrac{V_p d}{1.11\times10^{-5}} = \dfrac{(47.75)(1.667)}{1.11\times10^{-5}} = 6.42\times10^{6}$

e/d = RELATIVE ROUGHNESS = $.000005/1.667 = .000003$

f = FRICTION FACTOR = $.0088$

L_c = CASING LENGTH = $1.667'$

$HL = f\left(\dfrac{L_c}{d}\right)\left(\dfrac{V_p^2}{2g}\right) = (.0088)\left(\dfrac{1.667}{1.667}\right)\dfrac{(47.75)^2}{2(32.2)} = .3116'$

$k = \dfrac{HL_c}{Q_{nom}^2} = \dfrac{.3116}{(100)^2} = .0000312$

$HL_c = .0000312\ Q^2$

$$H_{RO} = HL_J + HL_I + HL_e - H_o$$

$$HL_J = \frac{V_J^2}{2g}$$

$$V_J = Q/A_J$$

$$A_J = A_p = 2.0942\ ft^2$$

$$HL_J = \frac{Q^2}{(2.0942)^2(2)(32.2)} = .00354\ Q^2$$

$$\left.\begin{aligned} HL_I &= .000234\ Q^2 \\ HL_e &= .0000312\ Q^2 \end{aligned}\right\}\ \text{DERIVATION \#1}$$

$$H_o = (RPR)\frac{V_o^2}{2g}$$

$$RPR = .70$$

$$H_o = \frac{(.70)\,V_o^2}{2(32.2)} = .0109\ V_o^2$$

$$H_{RO} = .00354\ Q^2 + .000234\ Q^2 + .0000312\ Q^2 - .0109\ V_o^2$$

$$= .00381\ Q^2 - .0109\ V_o^2$$

EXPANDED AREA

r	c	T.H.	$f(A_x)$
2	10.70	½	5.35
3	12.25	1	12.25
4	13.70	1	13.70
5	14.94	1	14.94
6	16.00	1	16.00
7	16.90	1	16.90
8	17.50	1	17.50
9	17.85	1	17.85
10	18.00	½	9.00
			123.49

$A_x = (3)(1)(123.49) = 370.47 \text{ in}^2$

$A_o = (.785)(20)^2 = 314 \text{ in}^2$

$EAR = \dfrac{370.47}{314} = 1.18$

PROJECTED AREA

r	c	α	[illegible]	T.M.	[illegible]
2	10.70	57.86	5.69	½	2.84
3	12.25	46.70	8.40	1	8.40
4	13.70	38.51	10.72	1	10.72
[illegible]	[illegible]	32.48	12.60	1	[illegible]
6	[illegible]	27.95	14.13	1	[illegible]
[illegible]	[illegible]	24.45	15.35	1	[illegible]
[illegible]	[illegible]	21.70	16.26	1	[illegible]
[illegible]	[illegible]	19.48	16.83	1	[illegible]
[illegible]	[illegible]	17.66	17.15	½	8.58
					105.74

$\alpha = \text{ARC TAN}\left(\dfrac{P}{2\pi}\right)/r$

$= \text{ARC TAN}\left(\dfrac{20}{2\pi}\right)/r$

$= \text{ARC TAN } 3.1831/r$

$A_p = (3)(1)(105.74) = 317.22 \text{ in}^2 =$

$A_o = 314 \text{ in}^2$

$PAR = \dfrac{317.22}{314} = 1.01$

STRUCTURAL ANALYSES

- PROPELLER
- SHAFT
- CASING

PROPELLER STRESS ... LIMITS

R = PROP RADIUS = 10"

Z = NO. OF BLADES = 3

P = PITCH = 20"

T_p = PROP. THRUST = $\rho g H_o A_p = (64.4)(20.9)(2.0942) = 2819^{\#}$

N = PROP. SPEED = $\dfrac{60 Q_{cr}}{A_p d_{eye} D} = \dfrac{(60)(893)}{(2.0942)(.905)(1.667)}$ = 1696 RPM

Q' = PROP TORQUE = $63024 \dfrac{SHP}{N} = \dfrac{(63024)(264)}{1696}$ = 10554 IN$^{\#}$

20 MPH CAV. LIMIT

$a = \dfrac{2\pi R}{P} = \dfrac{2\pi \, 10}{20} = 3.1416$

$x = r/R$

$K = f(x)$ TABLE I CONOLLY

$A_1 = f(a, x)$ " II "

$A_2 = f(a, x)$ " ... "

$B_1 = f(a ...)$ ... "

$B_2 = f(a, ...)$ V "

$C_1 = f(...)$ "

c = ...

t = MAX ... THICK...

σ_R = SPANWISE BENDING STRESS $= \dfrac{RK}{Z c t^2}\left[A_1\left(\dfrac{2\pi R T}{P}\right) + A_2\left(\dfrac{Q'}{R}\right)\right]$

σ_θ = CHORDWISE BENDING STRESS $= \dfrac{RK}{Z c t^2}\left[B_1\left(\dfrac{2\pi R T}{P}\right) + B_2\left(\dfrac{Q'}{R}\right)\right]$

σ_c = CENTRIFUGAL STRESS $= \dfrac{2240 N^2 R^2 C_1}{10^{10}}$

$s_{T_{MAX}}$ = MAX. TENSION STRESS (FACE) = $\sigma_R + \sigma_c$

$s_{S_{MAX}}$ = MAX. SHEAR STRESS (FACE) = $\dfrac{s_T - \sigma_\theta}{2}$

PROPELLER STRESS CALCULATION

BENDING STRESSES

R	Z	P	T	Q'	a	x	K	A_1	A_2	c	t	σ_R	B_1	B_2	σ_Q
10.00	3	20.00	2819	[illegible]	[illegible]	.20	.1203	6.60	57.95	10.70	.725	8654	2.26	19.30	[illegible]
						.30	.1207	6.94	39.26	12.25	.638	8303	3.13	18.69	[illegible]
						.40	.1007	7.27	31.20	13.70	.550	7882	3.97	19.25	4493
						.50	.0720	7.86	27.63	14.94	.463	7417	4.79	19.98	[illegible]
						.60	.0437	8.74	27.00	16.00	.375	6856	5.63	20.93	[illegible]
						.70	.0214	10.00	27.95	16.90	.288	6008	6.58	22.20	[illegible]
						.80	.0075	11.61	30.10	17.50	.201	4759	7.65	23.60	[illegible]
						.90	.0013	12.53	30.32	17.85	.113	2718	8.44	24.74	[illegible]

CENTRIFUGAL STRESS

R	N	x	C_1	σ_C
10.00	1696	.20	16.6	1070
		.30	12.0	773
		.40	9.5	612
		.50	7.7	496
		.60	6.2	399
		.70	4.8	309
		.80	3.4	219
		.90	1.9	122

COMBINED STRESSES

x	σ_R	σ_Q	σ_C	S_{TMAX}	S_{SMAX}
.20	8654	2879	1070	9724	3422
.30	8303	3828	773	9076	2939
.40	7882	4493	612	8494	2000
.50	7417	4759	496	7913	1577
.60	6856	4658	399	7255	1298
.70	6008	4158	309	6317	1080
.80	4759	3284	219	4978	847
.90	2718	1917	122	2840	462

FLOTATION SHAFT [illegible]

TORSIONAL STRESS

Q' = TORSION MOM. = 10554 IN#

d = SHAFT DIAM = 2", r = 1.00"

J = POLAR MOM. OF INERTIA $= \left(\frac{\pi}{2}\right) r^4 = \left(\frac{\pi}{2}\right)(1)^4 = 1.5708$

s_s = TORSIONAL STRESS $= \frac{Q' r}{J} = \frac{(10554)(1)}{1.5708} = 6719$ psi

FACTOR OF SAFETY $= \frac{20000}{6719} = 2.98$ ON SHEAR YIELD (6061-T6)

WHIRLING FREQUENCY

w = WEIGHT PER UNIT LENGTH $= (.785)(2)^2(1)(.096) = .3014$ #/IN

l = DISTANCE BETWEEN SUPPORTS = 36"

I = MOM. OF INERTIA $= .049 d^4 = (.049)(2)^4 = .784$ IN^4

D = STATIC [illegible] DUE TO OWN WT.

$= .00542 \frac{wl^4}{EI} = \frac{(.00542)(.3014)(36)^4}{(10,200,000)(.784)} = .000343$" FREE - FIXED

f = WHIRLING [illegible] $= \frac{3.55}{D^{1/2}} = \frac{3.55}{(.000343)^{1/2}} = 192$ CPS.

$= 11,499$ RPM

N_{OES} = 169[illegible]

[illegible]

INLET CASING DESIGN PRESSURE

p = DESIGN PRESSURE

= EXTERNAL PRESSURE − INTERNAL PRESSURE

$= \left(H_{ATM} + H_L - H_{IS}\right)\frac{64}{144}$

$= (33.08 + 1 - 26.38)\left(\frac{64}{144}\right) = 3.42$ psi

NOTE: MINIMUM H_{IS} OCCURS DURING STATIC OPERATION AT CAV. LIMIT

INLET CASING STRESS

p = DESIGN PRESSURE = 3.42 psi

l = SPAN = 24"

w = UNIT WIDTH = 1"

M = BENDING MOM. IN PLATE $= \frac{p\,l^2 w}{12} = \frac{(3.42)(24)^2(1)}{12} = 164.16$ in-lb

t = PLATE THICKNESS = .375"

Z = SECTION MODULUS OF PLATE $= \frac{w t^2}{6} = \frac{(1)(.375)^2}{6} = .0234$ in³/in width

s = BENDING STRESS IN PLATE $= \frac{M}{Z} = \frac{164.16}{.0234} = 7015$ psi

ESTIMATED WEIGHTS

[illegible]

INLET CASING

$$A = \left[2\frac{(34.5)(24)}{2} + (20)(34.5) + 2\left(\frac{35}{360}\right)(.785)(48)^2 + \left(\frac{35}{360}\right)\pi(48)(20)\right]\frac{1}{144} = 15.0[illegible]$$

$t = .375''$, $w = (.375)(144)(.096) = 5.18\ \#/FT^2$

$W = (15.02)(5.18) = 77.80\#$

TRANSITION

$$A = \left[\frac{2(24+70)}{2} + 20\pi\right]\frac{10}{144} = 5.24\ FT^2$$

$t = .375''$, $w = 5.18\ \#/FT^2$

$W = (5.24)(5.18) = 27.14\#$

PROPELLER CASING

$A = 20\pi(20)/144 = 8.73\ FT^2$

$t = .375''$, w [illegible]

$W = (8.73)(5.18)$ [illegible]

[illegible]

[illegible] $4(4)(.375)(.71)(8) = 34.08\ IN^3$

$w = .096\ \#/IN^3$

$W = (34.08)(.096)$ [illegible]

[illegible] TUF[illegible]

Sta	d	a	T.M.	f(▽)
1	4.00	12.56	½	6.28
2	4.00	12.56	1	12.56
3	3.50	9.62	1	9.62
4	2.75	5.94	½	2.97
				31.43

$\nabla = 4(31.43) - 8(.785)(2.125)^2 - 4(.785)(2.75)^2 = 66.55\ IN^3$

$w = .096\ \#/IN^3$

$W = (66.55)(.096) = 6.39\#$

CASING TOTAL

$W = 77.80 + 27.14 + 45.20 + 3.27 + 6.39 = 160\#$ CONV. CONST.

$= (160)\left(\frac{1.50}{2.66}\right) = 90\#$ COMPOSITE

[illegible]

BLADE

x	c	t	a	T.H.	$f(\nabla)$
.20	10.70	.725	5.57	1/2	2.76
.30	12.25	.638	5.55	1	5.55
.40	13.70	.550	5.35	1	5.35
.50	14.94	.463	4.91	1	4.91
.60	16.00	.375	4.26	1	4.26
.70	16.90	.288	3.46	1	3.46
.80	17.50	.201	2.50	1	2.50
.90	17.85	.113	1.43	1	1.43
1.00	18.00	.080	1.02	1/2	.51
					30.73

$\nabla = 3(1)(30.73) = 92.19 \, IN^3$
$w = .28^{\#}/IN^3$
$W = (92.19)(.28) = 25.81^{\#}$

HUB

$\nabla = (10)(.785)(4)^2 - (6)(.785)(1.75)^2 - (4)(.785)(3.25)^2 = 78.01 \, IN^3$
$w = .28^{\#}/IN^3$
$W = (78.01)(.28) = 21.84^{\#}$

TOTAL

$W = 25.81$
$\underline{21.84}$

$47.65^{\#}$

SHAFTING WEIGHT

SHAFT

$$\ell = 68/12 = 5.67'$$
$$w = (.785)(2)^2(12)(.096) = 3.62\ \#/FT$$
$$W = (5.67)(3.62) = 21\ \#$$

MISC. (BEARINGS, HOUSINGS, SEALS)

$$W = 10^{\#}$$

TOTALS

$$W = 21 + 10 = 31^{\#}$$

[illegible]

INLET CASING

$$\nabla = \frac{20(34.5)(24)}{2} + \left(\frac{35}{360}\right)(.785)(48)^2(20) = 11797 \text{ IN}^3$$

TRANSITION

$$\nabla = \left[\frac{(24)(20) + (.785)(20)^2}{2}\right] 10 = 3970 \text{ IN}^3$$

PROPELLER CASING

$$\nabla = 20(.785)(20)^2 = 6280 \text{ IN}^3$$

TOTALS

$$\nabla_{T} = 11797 + 3970 + 6280 = 22047 \text{ IN}^3 = 12.76 \text{ FT}^3$$

$$w = 64 \text{ [illegible]}$$

$$W = ([illegible]) = 817^{\#}$$

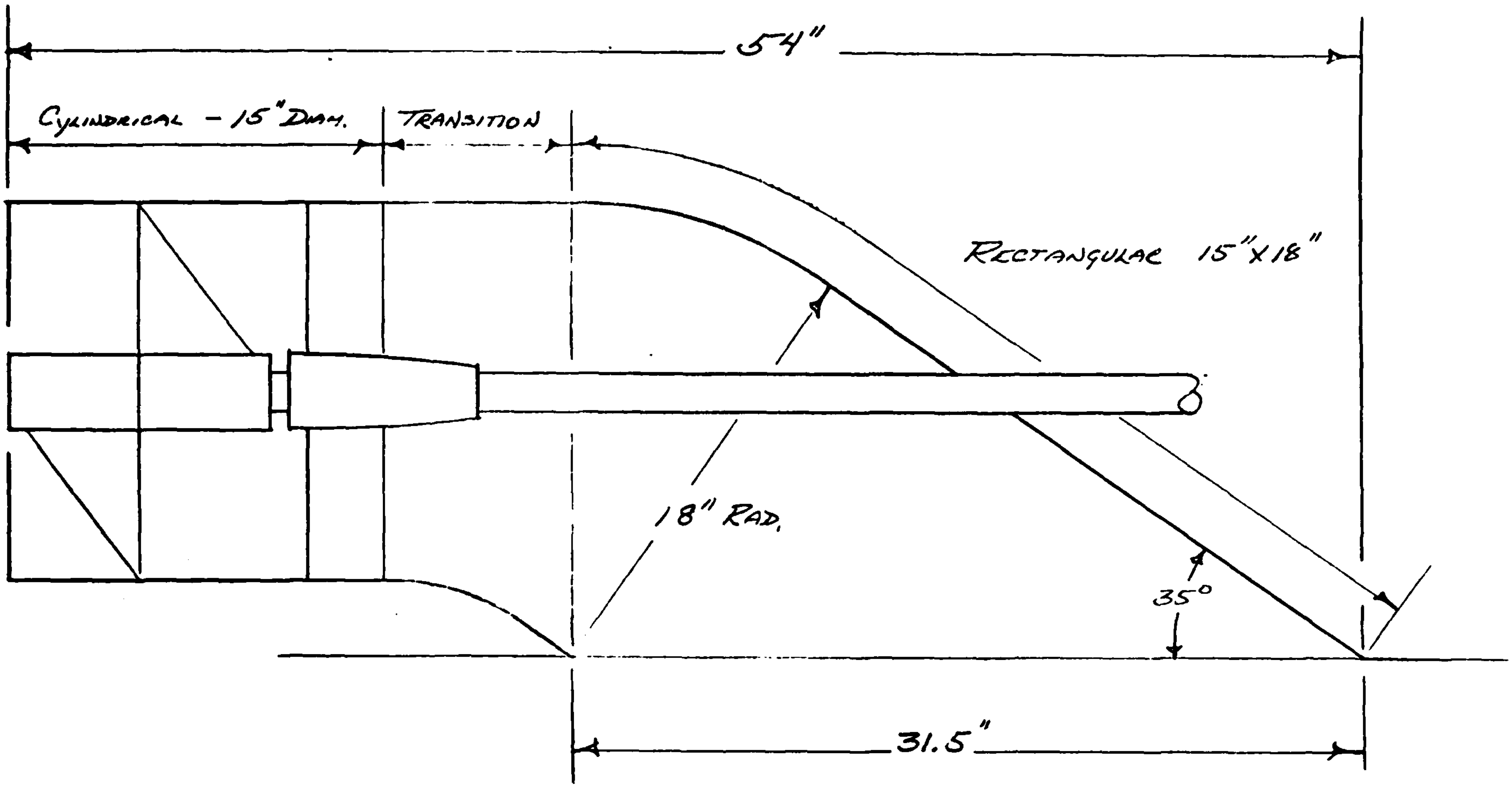
WEIGHT SUMMARY

	[illegible] CONST.		COMPOSITE CONST.	
	WT	MATL.	WT	MATL.
CASING	[illegible]	ALUM. ([illegible])	90	POLYESTER - GLASS CLOTH
PROPELLER	[illegible]	[illegible]	[illegible]	NICK AL-BR.
SHAFTING	[illegible]	ALUM. (6061-T6)	31	ALUM. (6061-T6)
DRY WT.	239#		169#	
WATER	[illegible]		817	
WET WT.	[illegible]		986#	

APPENDIX C

OBJECTIVES:

- Determine performance, weight and dimensional characteristics of a propulsion pump suitable for a multiple unit "tailgate" installation, using a 15-inch diameter impeller, in a high-speed (20 mph) amphibian.
- Use simple "propeller-in-tube" approach.
- Examine higher blade area ratios than available in existing propeller series data.



SKETCH OF 15 INCH DIAMETER PUMP

PERFORMANCE CHARACTERISTICS

- POWER LIMITS
- CAVITATION LIMITS
- SYSTEM PERFORMANCE
- STATIC OPERATION
- DERIVATIONS + REF. MATL.

POWER LIMIT

CALCULATION NOTES

D = PROP. DIAM. = $15'' = 1.25'$

A_P = PROP. DISC. AREA = $.785\left[(1.25)^2-(.25)^2\right] = 1.1775\ FT^2$

A_I = INLET AREA = $(1.50)(1.25) = 1.875\ FT^2$

J_{cyl} = ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE

J_e = ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE

$\frac{J_{cyl}}{J_e} = \frac{A_I}{A_P} = \frac{1.875}{1.1775} = 1.5924$

P/D = PROP. PITCH-DIAM. RATIO

$J_{cyl} = f\left(P/D, \frac{J_{cyl}}{J_e}\right) = .905$ (VON LAMEREN - FIG. 1)

$J_e = J_{cyl} / \left(\frac{J_{cyl}}{J_e}\right) = .905/1.5924 = .5683$

η_D = PROPULSION EFFICIENCY OF PROP.-TUBE COMB. $= f(J_e, P/D)$ (FIG. 2)

η_{RR} = PROP. RELATIVE ROTATIVE EFFICIENCY = .95 (GUESS - TUNNEL PROP.)

η_P = PUMP EFFIC[illegible] J_{cyl} [illegible] η_{RR}

SHP = [illegible]

Q = FLOW RATE [illegible] FT [illegible]

H_{PAV} = PUMP HEAD RISE AVAILABLE $\frac{550\ SHP\ \eta_P}{\rho g Q}$

Power Limit

Calculation

D	A_P	A_I	$\frac{J_{cyl}}{J_e}$	P/D	J_{cyl}	J_e	η_o	η_{ne}	η_p	SHP	Q	H_{avail}
1.25	1.1775	1.875	1.5924	1.00	.905	.5683	.51	.95	.77	500	40	82.20
											60	54.80
											80	41.10
										400	40	65.76
											60	43.84
											80	32.88
										300	40	49.32
											60	32.88
											80	24.66
										200	40	32.88
											60	21.92
											80	16.44
										100	40	16.44
											60	10.96
											80	8.22

CAVITATION [illegible]

CALCULATION NOTES

D = PROP. DIAM. = 15" = 1.25'

A_P = PROP. DISC AREA = $.785\left[(1.25)^2-(.25)^2\right]$ = 1.1775 FT^2

A_I = INLET AREA = (1.25)(1.50) = 1.875 FT^2

J_{CYL} = ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE

J_E = ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE

$$\frac{J_{CYL}}{J_E} = \frac{A_I}{A_P} = \frac{1.875}{1.1775} = 1.5924$$

P/D = PROP. PITCH-DIAM. RATIO

$J_{CYL} = f\left(P/D, \frac{J_{CYL}}{J_E}\right)$ (VAN LAMEREN - FIG. 1)

Q = FLOW RATE ~ FT^3/SEC

n = PROP. SPEED = $\dfrac{Q}{A_P J_{CYL} D}$

HL_I = INLET HEAD LOSS = $.000766\,Q^2$ (DERIVATION #1)

V_I = INLET VELOCITY = $\dfrac{Q}{A_I}$

$V_{.7R\,TAN}$ = TANGENTIAL VELOCITY, AT .7R [illegible] $\pi D n$

$V_{.7R}$ = TOTAL VELOCITY, AT .7 RAD. OF [illegible] $\sqrt{\;[illegible]\; V_{.7R\,TAN}^2}$

H_{ATM} = HEAD DUE TO [illegible]

H_Z = HEAD DUE TO ELEVATION

H_V = HEAD DUE TO VAP. PRESS.

CAVITATION LIMIT

CALCULATION NOTE - (CONT.)

V_0 = FREE STREAM VELOCITY = CRAFT SPEED

RPR = RAM PRESS. RECOVERY RATIO = .70 (JACUZZI - FIG. 3)

H_0 = RAM HEAD RECOVERY = $(RPR)\frac{V_0^2}{2g}$

H_{I_S} = INLET STATIC HEAD (ABOVE VAP. PRESS.) = $H_{ATM} + H_2 - H_V + H_0 - H_{LI} - \frac{V_I^2}{2g}$

P_{I_S} = INLET STATIC PRESS. (ABOVE VAP. PRESS.) = $\rho g H_{I_S}$

$\sigma_{.7R}$ = LOCAL CAV. NO. AT .7 RAD = $\frac{P_{I_S}}{\rho/2\, V_{.7R}^2}$

$\tau_{c_{MAX}}$ = PROP. LOAD COEF. AT CAV. LIMIT = $.7\,\sigma_{.7R}$ (GANN)

PAR = PROP. PROJECTED AREA RATIO

$H_{P_{CAV}}$ = PUMP HEAD RISE AT CAV. LIMIT = $\frac{(\tau_{c_{MAX}})(PAR)(V_{.7R})^2(\rho/2)}{\rho g}$

...TATION LIMIT

CALCULATION

I	Dp	Ds	Jss/Js	P/D	Jc	Q	n	HLs	VI	Vmod	Vc	VI²/2g	Havl	He	HL
[illegible]	1.1775	1.875	1.5924	1.00	.905	40	30.?3	1.23	21.33	[illegible]	[illegible]	7.06	33.06	3.00	1.00
						60	45.04	2.76	32.00	123.61	127.88	15.90			
						80	60.06	4.10	[illegible]	[illegible]	[illegible]	28.27			

Q	Vo	RPR	Ho	Hxs	P	σTR	τ	PAR	Hcal	PAR	Hcal
40	14.70	.70	2.35	[illegible]	1577	.2552	.1507	1.00	20.40	1.50	30.60
60				[illegible]	1209	.0739	.0517		13.13		19.70
80				[illegible]	274	.0094	.0066		2.78		4.47
40	22.05	.70	5.26	32.07	2065	.2441	.1989		22.45		33.68
60				21.70	1397	.0854	.0548		15.19		22.78
80				7.19	463	.0154	.0111		5.01		7.52
40	27.40	.70	[illegible]	[illegible]	2331	.3207	.3245		25.34		38.01
60				25.82	1663	.1017	.0712		15.08		27.12
80				11.31	728	.0250	.0135		7.90		11.85
40	36.25	.70	14.68	41.47	2671	.3674	.2572		29.03		43.54
60				31.10	2003	.1225	.0858		21.79		32.68
80				16.59	1068	.0367	.0257		11.60		17.40
40	0		[illegible]	26.79	1725	.2333	.1661				
60				[illegible]	1057	[illegible]	.0452				
80				7.91	23	.0042	.0027				

Vo	RPR	Ho	Hxs	P	σTR	τ	PAR	Hcav	PAR	Hcal	PAR	Hcal
14.70	.70	2.35	23.14	1577	.2552	.1507	2.00	40.80	2.50	51.00	3.00	61.20
			18.77	1209	.0739	.0517		26.26		32.82		39.39
			4.26	274	.0094	.0066		5.96		7.45		8.94
22.05	.70	5.26	32.07	2065	.2441	.1989		44.90		56.12		67.35
			21.70	1397	.0854	.0548		30.36		37.98		45.57
			7.19	463	.0154	.0111		10.02		12.52		15.03
27.40	.70	[illegible]	[illegible]	2331	.3207	.3245		50.68		63.35		76.02
			25.82	1663	.1017	.0712		36.16		45.20		54.24
			11.31	728	.0250	.0135		15.80		19.75		23.70
36.25	.70	14.68	41.47	2671	.3674	.2572		58.06		72.58		87.09
			31.10	2003	.1225	.0858		43.58		54.48		65.37
			16.59	1068	.0367	.0257		23.20		29.00		34.80
0		[illegible]	26.79	1725	.2333	.1661					3.00	[illegible]
			[illegible]	1057	[illegible]	.0452						[illegible]
			7.91	23	[illegible]	[illegible]						3.93

CALCULATION NOTES:

- REQUIRED PUMP HEAD RISE

  V_0 = CRAFT SPEED

  Q = FLOW RATE ~ FT^3/SEC

  $H_{PREQ} = .0121\,Q^2 - .0109\,V_0^2$ (DERIVATION # 2)

- ESTIMATED THRUST (POWER LIMIT)

  V_0 = CRAFT SPEED

  SHP = PUMP INPUT POWER

  Q_{EQ} = EQUILIBRIUM FLOWRATE @ $H_{PAVAIL} = H_{PREQ}$.

  A_J = JET AREA = A_P = 1.1775 FT^2

  V_J = JET VELOCITY = Q_{EQ}/A_J

  T = THRUST = $\rho Q_{EQ}(V_J - V_0)$

  $P.C.$ = PROPULSIVE COEF. = $\dfrac{T V_0}{550\,SHP}$

- ESTIMATED THRUST (CAVITATION LIMIT)

  V_0 = CRAFT SPEED

  Q_{EQ} = EQUILIBRIUM FLOW RATE @ $H_{PCAV} = H_{PREQ}$.

  A_J = JET AREA = A_P = 1.1775 FT^2

  V_J = JET VELOCITY = Q_{EQ}/A_J

  T = THRUST = $\rho Q_{EQ}(V_J - V_0)$

  H_{PEQ} = EQUILIBRIUM HEAD RISE @ $H_{PCAV} = H_{PREQ}$.

  η_P = PUMP EFFICIENCY = .77 (POWER LIMIT CALC.)

  SHP = PUMP INPUT POWER = $\dfrac{\rho g H_{PEQ} Q_{EQ}}{550\,\eta_P}$

  $P.C.$ = PROPULSIVE COEF. = $\dfrac{T V_0}{550\,SHP}$

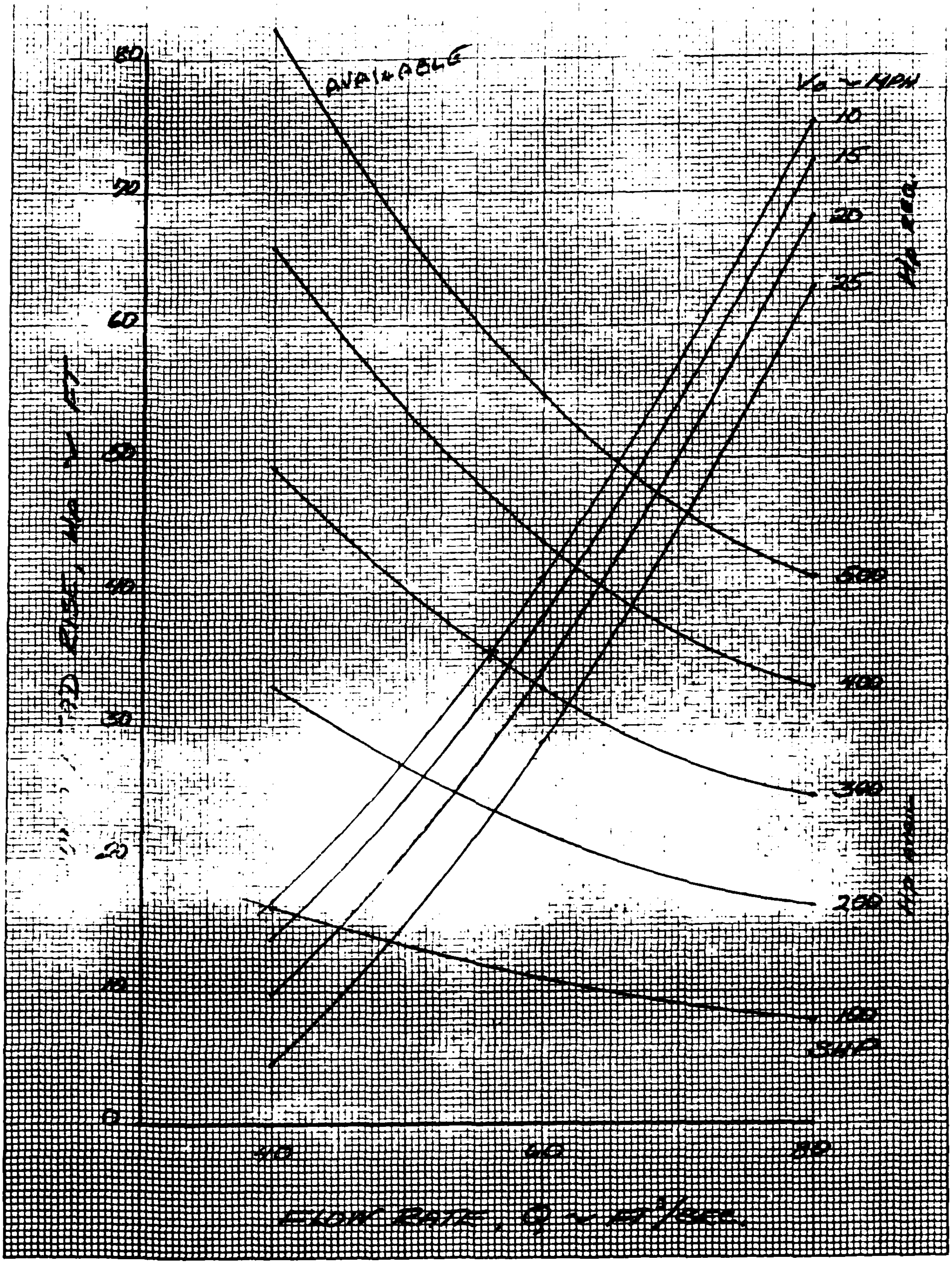
SYSTEM PERFORMANCE

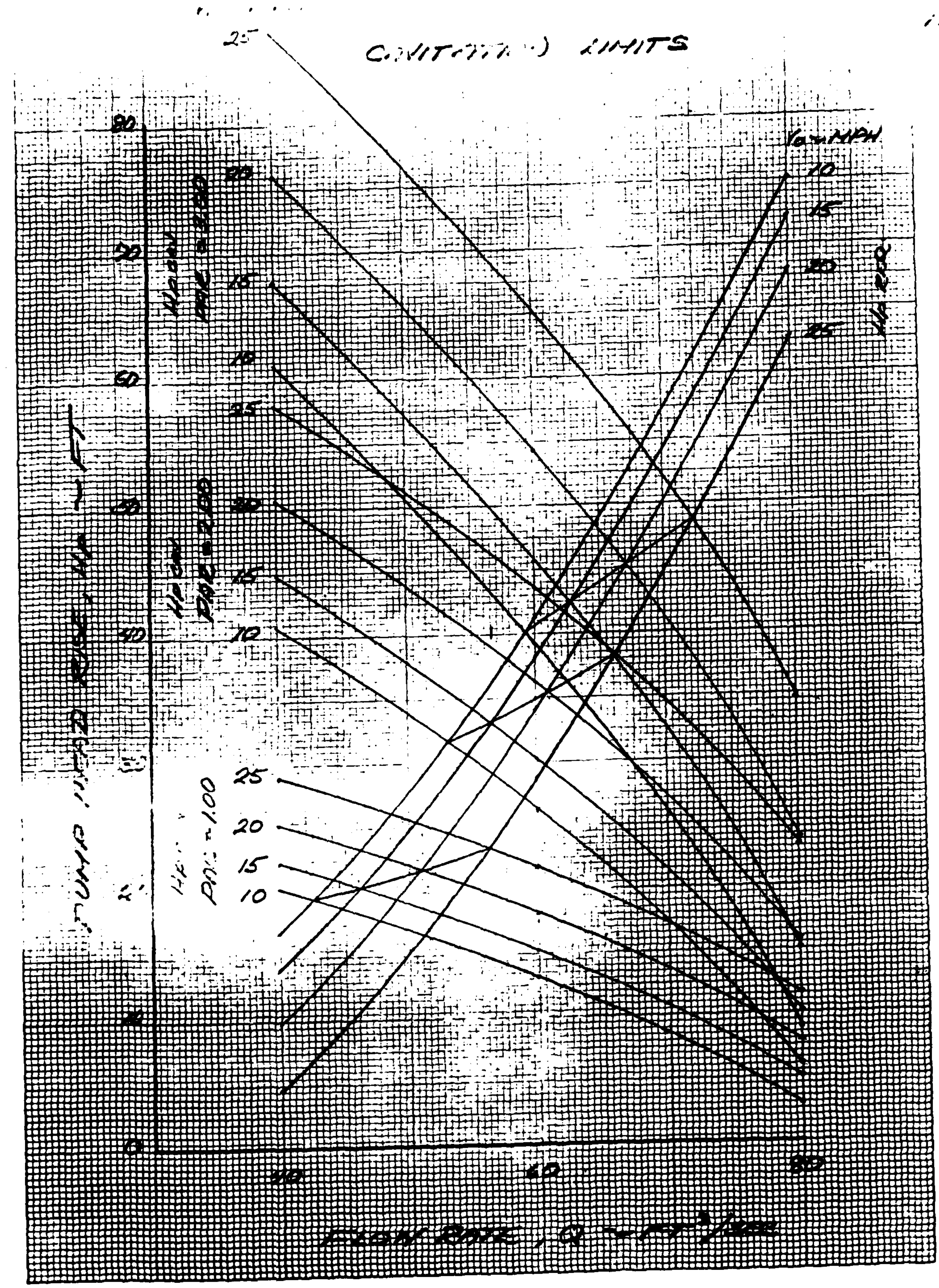
CALCULATIONS

- REQUIRED PUMP HEAD RISE

V_0 (ft/sec)	Q	H_{REQ}
14.70	40	17.00
	60	41.20
	80	75.08
22.05	40	14.06
	60	38.26
	80	72.14
29.40	40	9.94
	60	34.14
	80	68.02
36.75	40	4.64
	60	28.84
	80	62.72

POWER LIMITS





CROSS SECTION · 20 SQUARES TO INCH

CALCULATIONS (CONT.)

- ESTIMATED THRUST (POWER LIMIT)

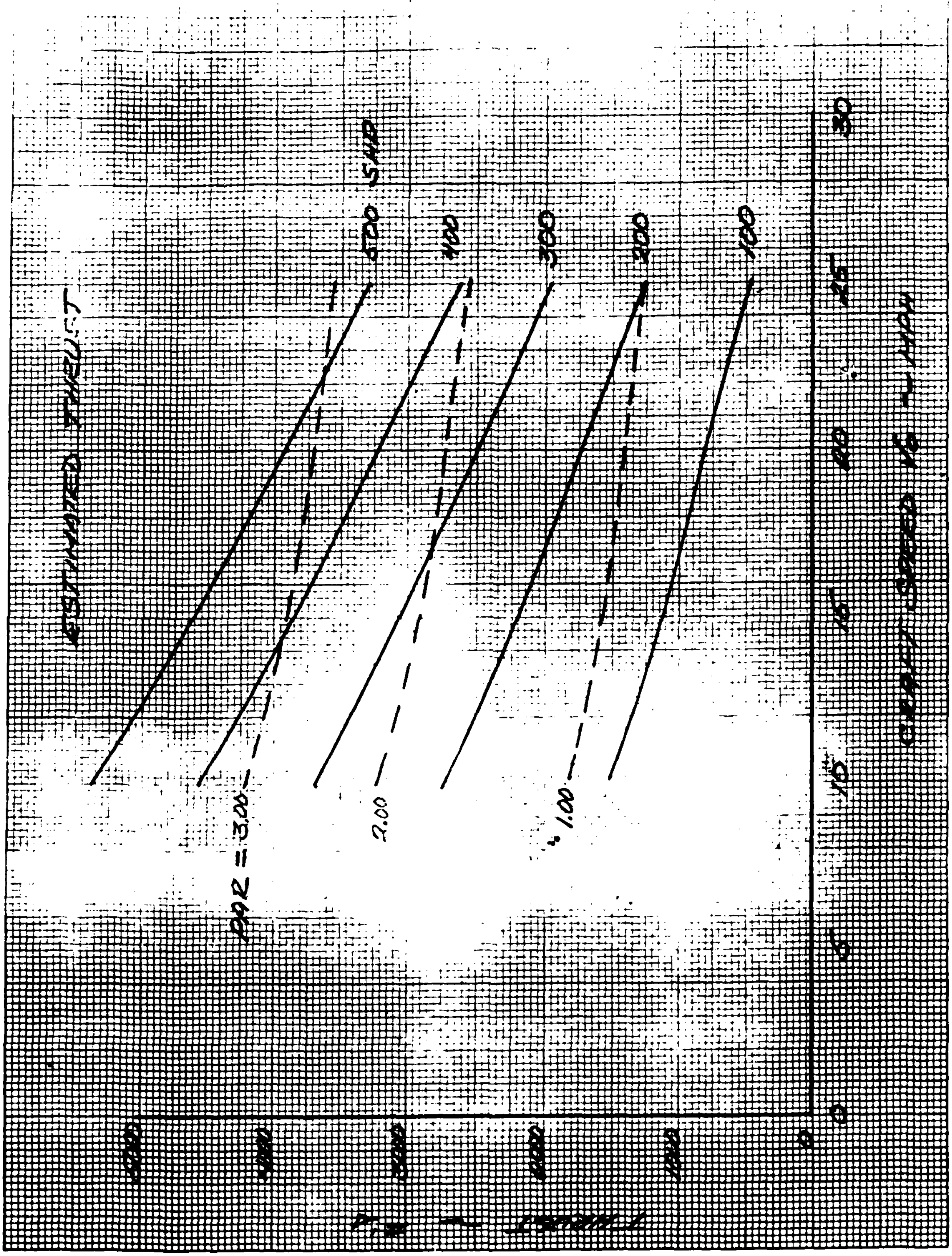
V_o (ft/sec)	SHP	Q_{EQ}	A_J	V_J	T	P.C.
14.70	500	65.6	1.1775	55.71	5381	.2876
	400	61.2		51.97	4562	.3048
	300	56.1		47.64	3696	.3293
	200	49.7		42.21	2734	.3654
	100	39.7		33.72	1510	.4036
22.05	500	66.7		56.65	4616	.3701
	400	62.4		52.99	3861	.3871
	300	57.4		48.75	3065	.4096
	200	51.3		43.57	2206	.4426
	100	41.8		35.50	1124	.4506
29.40	500	68.4		58.09	3925	.4196
	400	64.2		54.52	3225	.4310
	300	59.4		50.45	2501	.4456
	200	53.6		45.52	1728	.4618
	100	45.1		38.30	803	.4292
[illegible]	500	70.7		60.04	3293	.4401
	[illegible]	66.5		56.48	2624	.4383
	[illegible]	[illegible]		52.57	1959	.4363
	[illegible]	[illegible]		47.98	[illegible]	.4240
	[illegible]	[illegible]		41.53	467	[illegible]

[illegible]

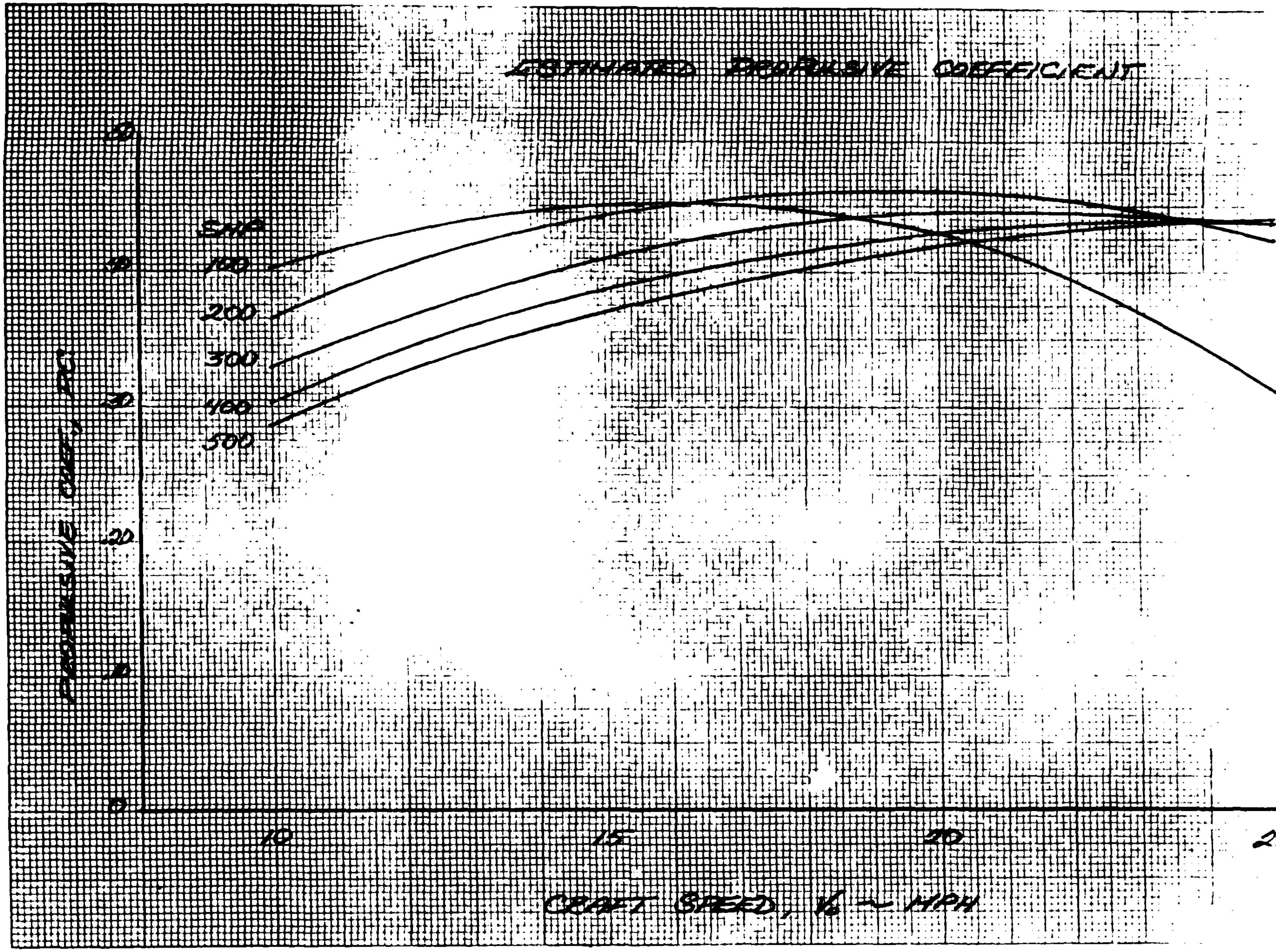
CALCULATIONS (CONT.)

- ESTIMATED THRUST (CAVITATION LIMIT)

PAR	V_o	Q_{EQ}	A_J	V_J	T	H_{PEQ}	η_p	SHP	P.C.
1.00	14.70	42.5	1.1775	36.09	1818	19.6	.77	127	.3826
	22.05	46.2		39.24	1588	20.4		143	.4452
	29.40	50.9		43.23	1408	21.8		169	.4453
	36.75	56.3		47.81	1245	23.3		199	.4180
2.00	14.70	53.2		45.18	3243	31.7		256	.3386
	22.05	56.3		47.81	2901	33.2		284	.4095
	29.40	60.9		51.72	2719	35.5		329	.4418
	36.75	66.1		52.14	2563	38.3		385	.4448
3.00	14.70	59.2		50.28	4213	40.2		362	.3111
	22.05	62.7		53.25	3912	42.4		404	.3852
	29.40	67.1		56.99	3703	45.3		462	.4284
	36.75	72.3		61.40	3564	48.7		535	.4451



22 116 CROSS SECTION 20 SQUARES TO INCH



22 116 · CROSS SECTION 20 SQUARES TO IN...

STATIC OP[illegible]ION

REQUIRED PUMP HEAD RISE

$H_{PREQ} = .0121\ Q^2$

Q	H_{PREQ}
40	19.36
60	43.56
80	77.44

CAVITATION LIMIT

(SEE CAVITATION LIMIT CALCS.)

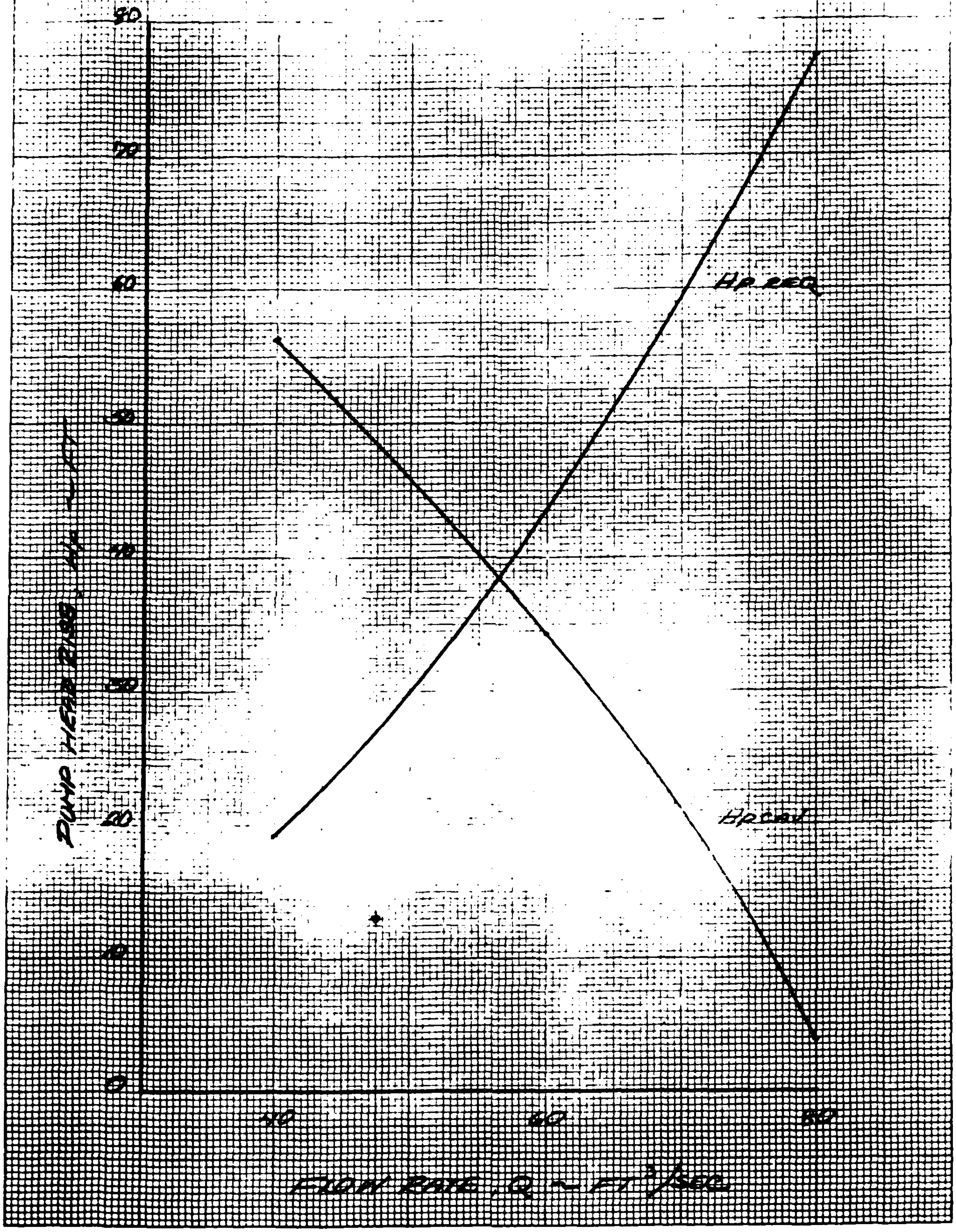
Q	H_{PCAV}
40	56.25
60	34.43
80	3.93

ESTIMATED THRUST

Q_{EQ}	A_J	V_J	T_{STAT}	H_{ATMOS}	H_L	[illegible]	[illegible]	V_I	$\frac{V_I^2}{2g}$	H_{IS}	[illegible]
56.4	1.1775	47.[illegible]	5403	3[illegible]	[illegible]	[illegible]	1.[illegible]	30.08	14.05	19.59	1262

USE TO DETERMINE INLET DESIGN PRESS. (STRUCT.)

STATIC OPERATION
CAVITATION LIMIT



DERIVA[illegible] & [illegible]

REF. MAT'L.

VON LAMEREN	FIG. 1	MODEL TESTS OF PROP. IN AXIAL CYL.
"	FIG. 2	
JACUZZI	FIG. 3	MODEL TEST OF INLET

DERIVATIONS

#1 ESTIMATED INLET & CASING LOSSES

#2 REQUIRED PUMP HEAD RISE

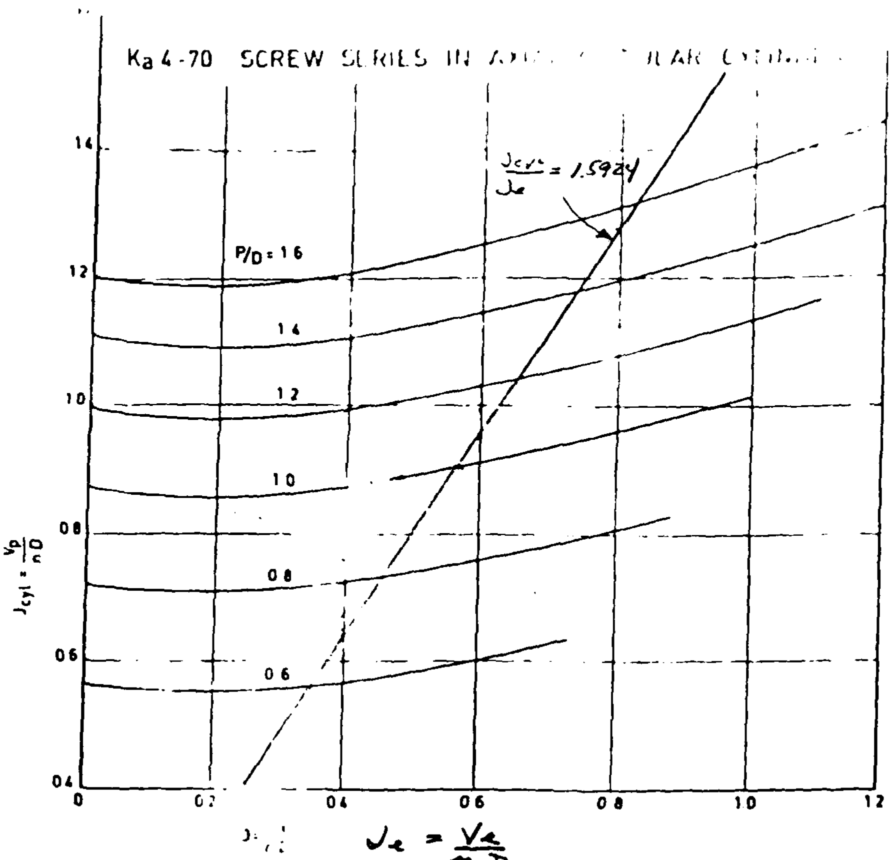


Fig. 1

Fig. 29 Relation between velocity of "screw + cylinder" combination and velocity in cylinder

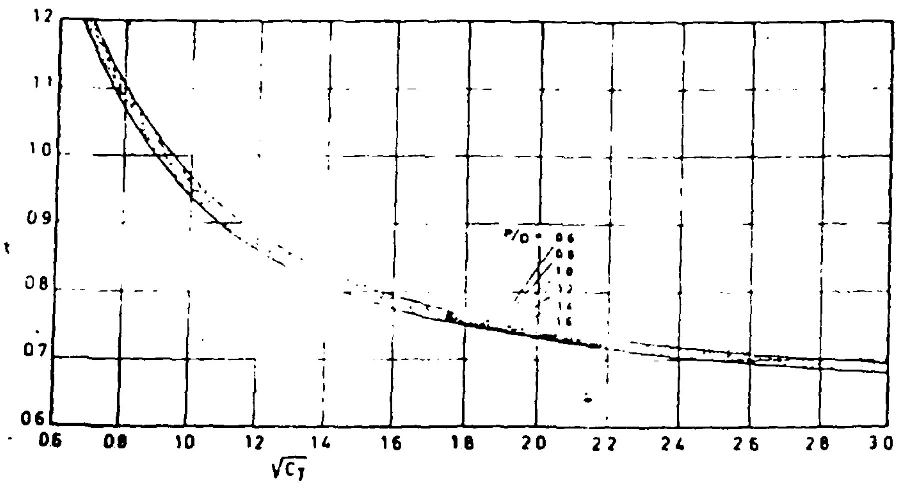


Fig. 30 Relation between thrust coefficient C_T and thrust ratio τ of nozzle no. 19a

figure have been obtained by substituting nozzles with different length-diameter ratios by systems of annular vortexes and calculating the induced velocities in the screw disk.

If the radial displacement of the streamlines is small, we can consider the streamlines as lying approximately on cylindrical planes. If internal friction and turbulence are neglected, the radial

Von Lameren

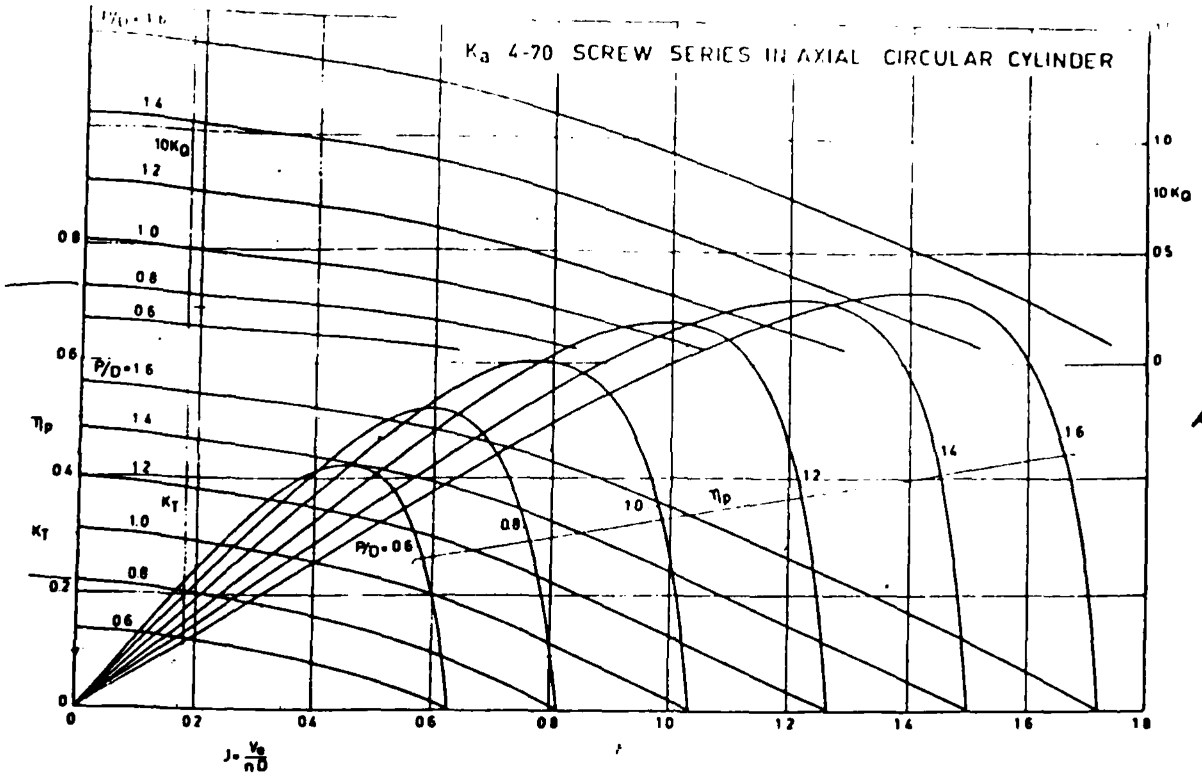


Fig. 2

Fig. 28 Results of open-water tests with Ka 4-70 screw series in an axial cylinder

been obtained from the experiments with the Ka 4-70 screw series in an axial circular cylinder and from the application of the momentum theorem.

From the comparison of the axial velocities obtained with these methods, we see that

1 The velocities agree reasonably well at high loadings of the ducted propeller system ($C_T > 1$).

2 The difference between the axial velocities becomes very large at low loadings ($C_T < 1$).

In regard to the second conclusion, the following remark may be made. From Fig. 13 it can be seen that the nozzle drag due to friction becomes substantial at low loadings of the ducted-propeller system. Then, it is no longer permitted to neglect the effect of friction on the force action between nozzle and fluid.

The design of a screw in a nozzle may now be carried out as follows:

With given thrust T or power P, intake velocity V_e, and number of revolutions n, the B_p and consequently the optimum diameter coefficient D can be determined with the aid of open-water test results of the nozzle considered, in combination with a systematic screw series (see, for instance, Fig. 24). In addition, the thrust coefficient C_T and the propeller thrust-total thrust ratio τ can be determined. With the aid of the experiments of the systematic screw series in the axial circular cylinder or using the momentum theorem, the axial velocity V_p in the way of the screw can be found. In addition, the mean axial velocity in the vicinity of the screw due to the nozzle action, U_N, and due to the screw action U_P, can be calculated.

The pressure difference created by the screw becomes

$$\Delta p = \frac{T_p}{\frac{\pi}{4}(D^2 - d_h^2)}$$

In order to avoid an excessive loading of the inner radii of the screw blades, the usual assumption for axial pumps that the head is constant for all radii is abandoned. The following radial $\Delta p(r/R)$ distribution is suggested for the screws in nozzle no. 19*a*:

$$\Delta p(r/R) = [4.88 - 4r/R]\cdot[r/R - 0.133]\Delta p$$

The radial distribution of the axial and tangential velocities at the screw may be approximated as follows:

A reasonable radial distribution of the axial velocities due to the nozzle action can be determined from Fig. 32. The results given in this

Von Lammeren

C-21

COMPARISON OF RECTANGULAR AND ELLIPTICAL INLET RAM RECOVERY VARIATIONS WITH INLET VELOCITY RATIO

(Laboratory Water Channel Test of 2-inch Eye Diameter Waterjet Inlet Models)

SYM	CONFIGURATION
□	RECTANGULAR
○	ELLIPTICAL – 0.3 IN. AFT LIP RADIUS

——— Estimated Performance of Jacuzzi Inlet Configuration

$$\eta_i = 1 - \frac{(P_{T_\infty} - \overline{P}_{i})}{q_\infty}$$

♦ Measured Performance of 28HJ Inlet

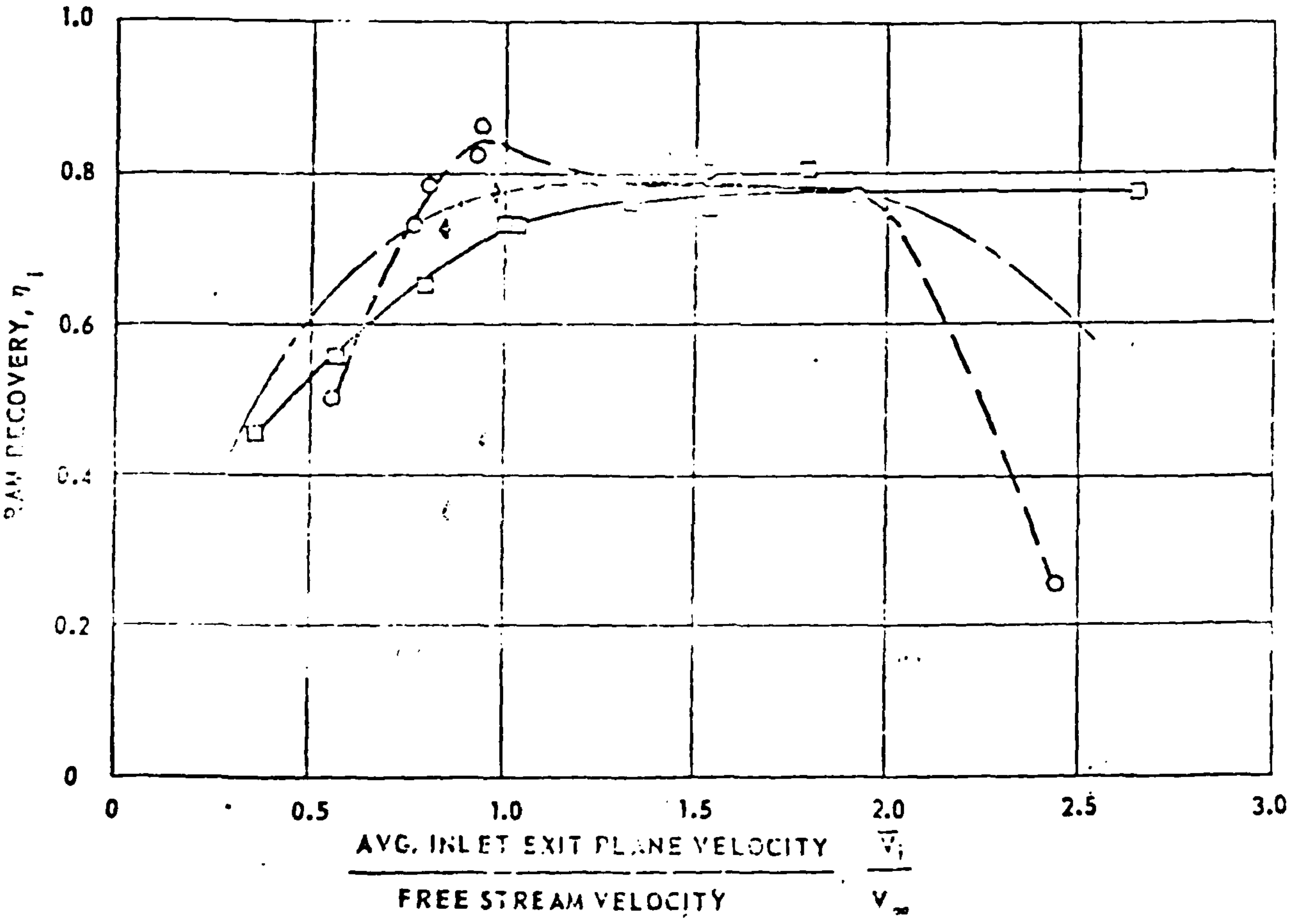


Fig. 3

[illegible] ([illegible])

<u>[illegible] FRICTION [illegible]</u>

Q = NOMINAL FLOW RATE = 60 FT^3/SEC

A_I = INLET AREA = $(1.25)(1.50) = 1.875$ FT^2

V_I = INLET VELOCITY = $Q/A_I = 60/1.875 = 32.00$ FT/SEC

d_e = EQUIV. INLET DIAM = $\dfrac{4A_I}{2(1.25+1.50)} = \dfrac{4(1.875)}{2(1.25+1.50)} = 1.36'$

$R_e = \dfrac{V_I d_e}{1.24\times10^{-5}} = \dfrac{(32)(1.36)}{1.24\times10^{-5}} = 3.51\times10^{6}$

ε/d_e = RELATIVE ROUGHNESS = .000004

f = FRICTION FACTOR = .0096

L_I = INTAKE LENGTH = $\sqrt{15.25^2-9^2} = 12.93'' = 1.08'$

L_B = EQUIV. LENGTH OF BEND = $\left(\frac{L}{D}\right)_B (d_e)\left(\frac{\alpha}{90}\right) = (36)(1.36)\left(\frac{35}{90}\right) = 19.04'$

L = TOTAL EQUIV. LENGTH = $L_I + L_B = 1.08+19.04 = 20.12'$

$HL = f\left(\frac{L}{d_e}\right)\left(\frac{V_I^2}{2g}\right) = (.0096)\left(\frac{20.12}{1.36}\right)\frac{(32)^2}{2(32.2)} = 2.258'$

<u>SHAFT</u>

C_D DRAG COEF. DUE TO CROSSFLOW = $1.1\sin^3\left(\frac{\alpha}{2}\right) = (1.1)\sin^3\left(\frac{35}{2}\right) = .0299$

V_I 32 FT/SEC

ℓ SHAFT LENGTH = 2'

d = SHAFT DIAM = 1.50" = .125'

D SHAFT "DRAG" = $C_D \frac{\rho}{2} V_I^2 \ell d = (.0299)(3/2)(32)^2(2)(.125) = 2.65^{\#}$

$HL = \dfrac{D}{\rho g A_I} = \dfrac{2.65}{2(32.2)(1.875)} = .0634'$

[illegible] (DERIVATION #1)

TRANSITION

A_T = CROSS SECTION AREA = $\frac{1.875 + 1.1775}{2}$ = 1.5263 FT^2

V_T = VELOCITY = Q/A_T = $60/1.5263$ = 39.31 FT/SEC

d_e = EQUIV. DIAM. = $\sqrt{\frac{1.5263}{.785}}$ = 1.39'

$R_e = \frac{V_T d_e}{1.24 \times 10^{-5}} = \frac{(39.31)(1.39)}{1.24 \times 10^{-5}} = 4.41 \times 10^6$

e/D = RELATIVE ROUGHNESS = .0000035

f = FRICTION FACTOR = .0093

L_T = TRANSITION LENGTH = 7.5" = .625'

$HL = f\left(\frac{L}{d_e}\right)\frac{V_T^2}{2(32.2)} = (.0093)\left(\frac{.625}{1.39}\right)\frac{(39.31)^2}{(2)(32.2)} = .100'$

BEARING TUBE

A_P = CROSS SECTION AREA = $(.96)(.785)(1.25)^2 = 1.1775$ FT^2

V_P = VELOCITY = Q/A_P = $60/1.1775$ = 50.96 FT/SEC

l = TUBE LENGTH = 1.25'

$R_e = \frac{V_P l}{1.24 \times 10^{-5}}$ $\frac{(50.96)(1.25)}{1.24 \times 10^{-5}} = 5.14 \times 10^6$

C_f = .00327

S = TUBE WETTED SURF = $(1.25)\pi(.25)$ = .98 FT^2

D = TUBE DRAG = $(C_f + .0008)(S)(\rho/2) V_P^2 = (.00327 + .0008)(.98)(2/2)(50.96)^2 = 10.36$#

$HL = \frac{D}{\rho g A_P} = \frac{10.36}{2(32.2)(1.1775)} = .137'$

[illegible] (CONTINUED)

STRUTS.

A_p = CROSS SECTION AREA = $1.1775 \, FT^2$

V_p = VELOCITY = $Q/A_p = 60/1.1775 = 50.96 \, FT/SEC$

c = STRUT CHORD = $3'' = .25'$

$R_e = \dfrac{V_p c}{1.24 \times 10^{-5}} = \dfrac{(50.96)(.25)}{1.24 \times 10^{-5}} = 1.03 \times 10^6$

$C_f = .00437$

t/c = STRUT THICKNESS RATIO $= .3125/3 = .1042$

$C_D = 2(C_f + .0008)(1 + 1.2\, t/c) = 2(.00437 + .0008)(1 + 1.2 \times .1042) = .0116$

S = STRUT PLANFORM AREA $= 4(.50)(.25) = .50 \, FT^2$

D = STRUT DRAG $= C_D \, S \, \frac{\rho}{2} V_p^2 = (.0116)(.50)(\frac{2}{2})(50.96)^2 = 15.06^{\#}$

$HL = \dfrac{D}{\rho g A_p} = \dfrac{15.06}{2(32.2)(1.1775)} = .199'$

TOTAL INLET LOSS

INTAKE FRICTION + BEND	2.258
SHAFT	.063
[illegible]NSITION	.100
[illegible]	.137
[illegible]	.199
[illegible]	2.757'

$k = \dfrac{HL_I}{Q_{NOM}} = \dfrac{2.757}{(60)^2} = .000766$

$HL_I = .000766 \, Q^2$

Casing

Q = NOMINAL FLOW RATE = 60 FT^3/SEC

A_P = 1.1775 FT^2

V_P = Q/A_P = 60/1.1775 = 50.96 FT/SEC

d = CASING DIAM = 1.25'

$R_e = \dfrac{V_P d}{1.24 \times 10^{-5}} = \dfrac{(50.96)(1.25)}{1.24 \times 10^{-5}} = 5.14 \times 10^6$

e/d = RELATIVE ROUGHNESS = .000004

f = FRICTION FACTOR = .009

L_c = CASING LENGTH = 1.25'

$HL = f\left(\dfrac{L_c}{d}\right)\left(\dfrac{V_P^2}{2g}\right) = (.009)\left(\dfrac{1.25}{1.25}\right)\dfrac{(50.96)^2}{2(32.2)} = .363'$

$k = \dfrac{HL_c}{(Q_{nom})^2} = \dfrac{.363}{(60)^2} = .000101$

$HL_c = .000101\ Q^2$

$$H_{\dots} = HL_J + HL_I + \dots - H_O$$

$$HL_J = \frac{V_J^2}{2g} \qquad V_J = \frac{Q}{A_J} \qquad A_J = A_f = 1.1225\ FT^2$$

$$HL_J = \frac{Q^2}{(1.1225)^2(2)(32.2)} = .0112\ Q^2$$

$$HL_I = .000766\ Q^2$$

(DERIVATION #1)

$$HL_C = .000101\ Q^2$$

$$H_O = (RPR)\frac{V_O^2}{2g} \qquad RPR = .70$$

$$H_O = \frac{(.70)\ V_O^2}{2(32.2)} = .0109\ V_O^2$$

$$H_{PRED} = .0112\ Q^2 + .000766\ Q^2 + .000101\ Q^2 - .0109\ V_O^2$$

$$= .0121\ Q^2 - .0109\ V_O^2$$

STRUCTURAL ANALYSES

- SHAFT
- CASING

Propeller Shaft

Torsional Stress

N = PROP. SPEED = $\frac{60 Q_{EQ}}{A_P J_{C,-} D}$ = $\frac{(60)(67.1)}{(1.1775)(.905)(1.25)}$ = 3022 RPM

SHP = 462

Q' = PROP TORQUE = $63024 \frac{SHP}{N} = (63024)\left(\frac{462}{3022}\right)$ = 9635 IN#

20 mph CAT. LIMIT PAR = 3.00

d = SHAFT DIAM. = 1.50"; r = .75"

J = POLAR MOM. OF INERTIA = $\frac{\pi}{2} r^4 = \left(\frac{\pi}{2}\right)(.75)^4 = .4970$

S_t = TORSIONAL STRESS = $\frac{Q' r}{J} = \frac{(9635)(.75)}{(.4970)}$ = 14,540 psi

FACTOR OF SAFETY = $\frac{20,000}{14540}$ = 4.81 ON SHEAR YIELD (AQUAMET 22)

Whirling Frequency

w = WEIGHT PER UNIT LENGTH = $(.785)(1.5)^2(1)(.28)$ = .4946 #/IN

l = DISTANCE BETWEEN SUPPORTS = 29"

I = MOM. OF INERTIA = $.049 d^4 = (.049)(1.5)^4$ = .2481 IN⁴

D = STATIC DEFLECTION DUE TO OWN WT

= $.00542 \frac{w l^4}{EI}$ = [illegible] = .000205" FREE - FIXED

f = WHIRLING FREQUENCY = $\frac{3.55}{D^{1/2}}$ = [illegible] = 248 cps

= 14,874 RPM

N_{DES} = 3022 RPM

Casing Structure

Inlet Casing Design Pressure

p = DESIGN PRESSURE

EXTERNAL PRESSURE − INTERNAL PRESSURE

$$\left(H_{ATM.} + H_{2} - H_{I_3}\right)\frac{64}{144}$$

$$= \left(33.08 + 3.00 - 19.59\right)\frac{64}{144} = 7.33\ psi$$

NOTE: MINIMUM H_{I_3} OCCURS DURING STATIC OPERATION AT CAV. LIMIT

Inlet Casing Stress

p = DESIGN PRESSURE = 7.33 psi

l = SPAN = 18"

w = UNIT WIDTH = 1"

M = BENDING MOM. IN PLATE $= \frac{p l^2 w}{12} = \frac{(7.33)(18)^2(1)}{12} = 197.91$ IN#/IN WIDTH

t = PLATE THICKNESS = [illegible]

Z = SECTION MOD[illegible] = [illegible] .0163 IN³/IN WIDTH

s = [illegible] $= \frac{197.91}{.0163} =$ [illegible] psi

ESTIMATED WEIGHTS

Casing Weight

Inlet Casing

$$A = \left[2\frac{(25.85)(18)}{2} + (15)(25.85) + 2\left(\frac{35}{360}\right)(.785)(36)^2 + \left(\frac{35}{360}\right)\pi(36)(15)\right]\frac{1}{144} = 8.44\ FT^2$$

$$t = .3125'', \quad w = (.3125)(144)(.096) = 4.32\ \#/FT^2$$

$$W = (8.44)(4.32) = 36.46\#$$

Transition

$$A = \left[\frac{2(15+18)}{2} + 15\pi\right]\frac{2.5}{144} = 2.95\ FT^2$$

$$t = .3125'', \quad w = 4.32\ \#/FT^2$$

$$W = (2.95)(4.32) = 12.73\#$$

Propeller Casing

$$A = 15\pi(15)/144 = 4.91\ FT^2$$

$$t = .3125'', \quad w = 4.32\ \#/FT^2$$

$$W = (4.91)(4.32) = 21.21\#$$

Struts

$$\nabla = 4(3)(.3125)(.71)(6) = 15.98\ IN^3$$

$$w = .096\ \#/IN^3$$

$$W = (15.98)(.096) = 1.53\#$$

Bearing Tube

STA	l	a	T.M.	$f(\nabla)$
1	3.00	7.06	½	3.53
2	3.00	7.06	1	7.06
3	2.625	5.41	1	5.41
4	2.06	3.34	½	1.67
				17.67

$$\nabla = 3(17.67) - 6(.785)(2)^2 - 3(.785)(1.75)^2 = 26.96\ IN^3$$

$$w = .096\ \#/IN^3$$

$$W = (26.96)(.096) = 2.59\#$$

Casing Total

$$W = 36.46 + 12.73 + 21.21 + 1.53 + 2.59 = 75\# \quad \text{CONV. CONST.}$$

$$= (75)\left(\frac{1.50}{2.66}\right) = 42\# \quad \text{COMPOSITE CONST.}$$

PROPELLER WT.

$$W = (47.65)\left(\frac{15}{20}\right)^3\left(\frac{PAR}{1.01}\right) = 19.90(PAR)$$

$$PAR = 3.00$$

$$= (19.90)(3) \cdot \; 60^{\#} \quad \text{CONV. CONST.}$$

$$= (60)\left(\frac{1.50}{2.66}\right) = 34^{\#} \quad \text{COMPOSITE CONST.}$$

SHAFTING W[illegible]

SHAFT

ℓ = 54" = 4.50 FT

w = (.785)(1.50)^2(12)(.28) = 5.93#/FT.

W = (4.50)(5.93) = 26.71#

MISC. (BEARINGS, SEALS, HOUSING)

W = 5#

SHAFTING TOTAL

W = 26.71 + 5 = 32#

Water Weight

Inlet Casing

$$\nabla = (15)\frac{(25.65)(18)}{2} + \left(\frac{35}{360}\right)(.785)(36)^2(15) = 4974 \text{ in}^3$$

Transition

$$\nabla = \left[\frac{(18)(20) + .785(15)^2}{2}\right](7.5) = 2012 \text{ in}^3$$

Propeller Casing

$$\nabla = (15)(.785)(15)^2 = 2649 \text{ in}^3$$

Totals

$$\nabla = 4974 + 2012 + 2649 = 9635 \text{ in}^3 = 5.58 \text{ ft}^3$$

$$w = 64\ \#/\text{ft}^3$$

$$W = (5.58)(64) = 357\#$$

WEIGHT SUMMARY

	Conventional Const.		Composite Const.	
	Wt	Matl.	Wt	Matl.
Casing	75	Alum. (5086-H116)	42	Polyester - Glass Cloth
Propeller	60	Nickel Al. Br.	34.	Polycarb. - Glass
Shafting	32	Aquamet 22	32	Aquamet 22
Dry Weight	167#		108#	
Water	357		357	
Wet Weight	524#		465#	

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